TREATISE ON STEAM,

AND THE

USE OF THE INDICATOR.

AN

ELEMENTARY TREATISE

ON

STEAM,

AND

THE USE OF THE INDICATOR.

BY

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PREFACE.

This small work is written for those who are beginning their studies in connection with engines, as the author ventures to think that there is at present no one work which contains in an elementary form all the subjects here treated of, and a reference to those large works which do contain them is frequently neither convenient nor possible.

The authorities which the author has chiefly consulted are Regnault's great work, Tyndall on 'Heat as a Mode of Motion,' Balfour Stewart's 'Elementary Treatise on Heat,' Clerk-Maxwell's 'Theory of Heat,' Williamson's 'Chemistry' and Miller's 'Chemistry,' Dr. Percy's 'Metallurgy,' and the Rev. Robert Dixon's 'Treatise on Heat.'

The author is much indebted to many gentlemen for the assistance and information which they have given him, and which will, where possible, be found to be acknowledged in the text. To Mr. Knight, manager of Messrs. John Penn and Sons' Works, the author feels that he is much indebted.

J. C. G.

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AN

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CHAPTER I.

THE INDICATOR CARD.

THE main object of an indicator is to ascertain the power which an engine is exerting. It does not do this exactly, but with sufficient accuracy for all practical purposes. The indicator is also useful in other respects.

In principle it is nothing more than an instrument for registering the varying steam pressures in the cylinder during a complete revolution of the shaft, or, if there is no shaft, during a complete reciprocation of the piston.

Putting aside for the present the mechanical arrangements as it is now made, and considering it in its simplest form, it is merely a small piston working in a cylinder with considerable clearance, carrying a pencil at the end of its piston rod. One end of this small cylinder is placed, at pleasure, in connection with either end of the main cylinder (that is, the cylinder of which it is desired to learn the horse-power), by means of a cock and pipes, and the other end of the indicator cylinder is in free communication with the air, by means of the loose stuffing box through which the piston rod of the indicator moves, and further by means of a hole drilled in the cylinder cover of the indicator, so that if steam goes into the main cylinder a portion of it may be admitted directly to the bottom side of the indicator piston, while upon the other side the air presses continually with whatever the barometric pressure may be at the time.

A spiral spring is attached to the cover of the indicator cylinder at one end, and to the indicator piston itself at the other. This spring regulates the movements of the piston, and as the steam is at a greater or less pressure, so the spring is more or less compressed.

Supposing now that the piston of the engine is at one end of the stroke, and about to begin a fresh stroke, and that steam is admitted to the cylinder for this purpose, the indicator spring will be compressed by the steam pressure under it, and the amount to which the indicator piston rises is a measure of the steam pressure; for example, supposing that the spring compresses $\frac{1}{8}$th inch for every pound on it, then, if the steam pressure is 20 lbs., the piston will rise $2\frac{1}{2}$ inches. As the piston of the engine travels forward on its stroke, the steam pressure begins to diminish, and becomes less and less able to compress the indicator spring, and consequently the indicator piston continually falls. In order to register these continually varying pressures, a piece of paper is held on a small cylinder or barrel in front of the pencil on the indicator piston, and as the main piston moves backwards and forwards the barrel of the indicator partially rotates backwards and forwards; and the curved line traced by the pencil moving vertically up and down, on the paper moving at right angles to the up and down movement of the pencil, is called an indicator card or diagram. The diagram is nothing more than a register of the varying pressures in the cylinder as the piston moves to and fro.

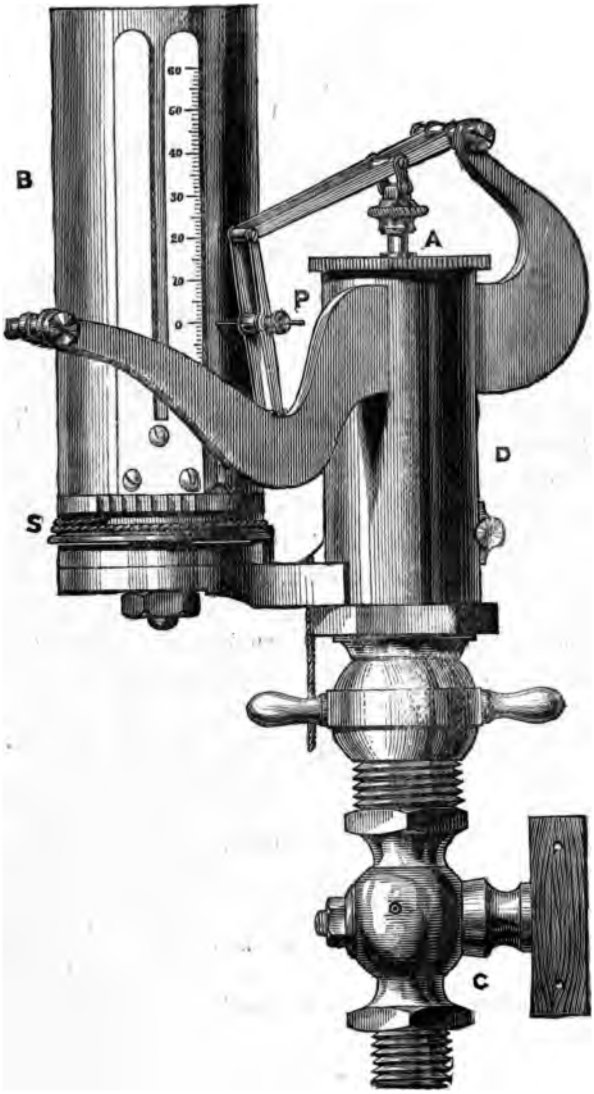
Fig. 1 is a drawing of the best form of indicator * now in use, as made by Messrs. Elliott Brothers, of the Strand. A is the piston rod, P the pencil, which is not attached directly to the piston rod, but by means of a parallel motion, so that it moves vertically, but travels through a considerably greater distance than the piston †; B is the barrel on which the paper is stretched; C is a cock, the use of which will be explained subsequently. It is screwed into a branch of another cock, by means of which latter, connection can be made at pleasure with either end of the cylinder. The indicator cylinder and spring are inside the casing D. S is the string which gives motion to the barrel B.

* Invented by Mr. Richards.

† Four times the distance travelled by the piston.

The other end of the string is fastened to some moving part of the engine, whose movements correspond with those of the piston, but

FIG. 1.



are smaller. A movement of four inches of the string gives a very good diagram.

If there is no part of the engine which can be made use of, such as the air or circulating pump levers, into which a stud can be screwed, some special appliance must be used for reducing the movement of the piston.

Below the cock marked C, another, but a two-way cock, comes, which is connected by means of pipes with both ends of the cylinder. By means of this second cock the indicator may be placed in communication with either end of the cylinder at pleasure, or may be shut off from both. The cock C is used in the following manner: Its plug being drilled three ways, $\vdash$, the indicator may by it be shut off from the cylinder and connected by means of the small hole in one side of the shell with the air, or it may be shut off from the air and connected with the cock below, and therefore with either end of the cylinder. When the barrel is in movement backwards and forwards, the plug is so turned that the indicator is placed in connection with the open air, and the spring having the air both on the top and bottom, is neither compressed nor extended. At this moment the pencil is pressed against the paper, where it rules a straight line, called the atmospheric line. It is important to remember this connection between the position of the pencil at the atmospheric line and the condition in which the spring then is.

We must now see what use can be made of this diagram or register of pressures. The connection between a curved figure and the power exerted by the engine is not at first sight apparent; and before showing what it is, it is necessary for me to endeavour to clear away all misunderstanding as to what is a true measure of power exerted. Without a most clear and definite conception of what constitutes a mechanical expenditure of energy, or, in other words, what is the measure of work done, it is impossible to form any notion either of what is meant by economical use of steam, or of the connection between the indicator diagram and the indicated horse-power.

The simplest example of an expenditure of power, and also the commonest, is that of a weight raised from the ground. If one pound has been raised one foot high, just half the energy has been required which would be required to raise two pounds one foot high. This is so simple a conception as not to require further

explanation. A little consideration will show that generally speaking the energy required to lift any weight to any height may be said to be equal to a certain number of pounds raised one foot high, or what is just the same thing, one pound raised a certain number of feet high, or as it is generally stated for the sake of shortness, the energy expended is equal to a certain number of foot-pounds.

It is a general law in mechanics that when energy or power is expended some resistance has been overcome through some distance, and what is really done in raising a weight is to overcome the attraction of the earth, or gravity, through a certain distance.

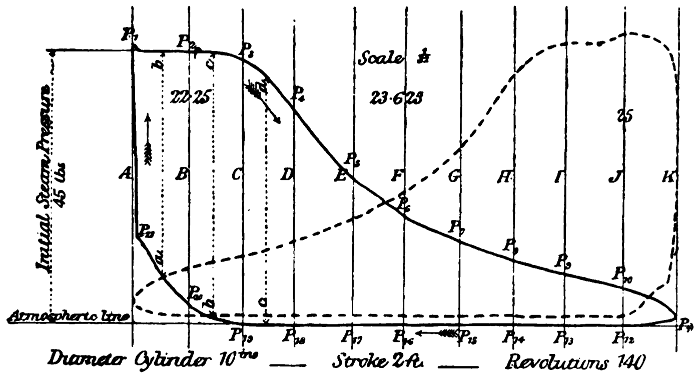
If we had overcome any other resistance than the attraction of gravity, as, for instance, compressing a spring, we might, in just the same way, say the expenditure of energy was equal to that required to lift a certain number of pounds through a certain height. We may make the attraction of gravity a general standard of resistance, and whenever any resistance is overcome, we may refer it to this standard. One pound raised one foot high may be taken as our standard unit of work done.

An engine at work overcomes some resistance, either propelling a vessel, or pulling a train, or driving machinery; and the amount of energy expended by the engine in overcoming this resistance through a certain distance is equivalent to a certain number of pounds raised through a certain number of feet.

To ascertain what power an engine is exerting, it appears the simplest way is to find out how many pounds weight it raises in a minute, and through how many feet it raises them; I use a minute as a convenient unit of time, and because it is the unit generally adopted. With this in view, take as an example the card in Fig. 2, and divide it into ten equal spaces. The distance from A to B is one-tenth of the whole length of the indicator card, and during the time the card travelled horizontally from A to B the piston of the engine travelled one-tenth of its stroke; while the card travelled from B to C the piston of the engine travelled another one-tenth of its stroke, and when the engine had travelled its whole stroke the card would have travelled from A to K, and so backwards, on the return stroke. It is not a matter of any importance what the

length A K is when compared with the stroke of the engine, and for convenience A K is usually made about 4 inches, excepting in engines running at high revolutions. All that we care about

FIG. 2.



is, that when the engine has moved through one-tenth of its stroke, the card shall have done so also, and that the motions go on corresponding in this way throughout the stroke. Then we have only to look at the indicator card to see what pressure of steam there was in the cylinder at any part of the stroke. In this particular case a $\frac{1}{32}$nd spring has been put in the indicator, which means that for every 1 lb. pressure on the square inch of the piston the pencil of the indicator will rise $\frac{1}{32}$nd inch. If we have 45 lbs. boiler pressure, the pencil will rise 1·4 inch as soon as the steam is admitted up to the point P_1; then, as the engine and card move, the pencil, still held up by the steam, moves to P_2, then to P_3. Somewhere about this point the steam is cut off, then the steam pressure falls as the piston moves on, and the pressure can no longer compress the spring so much, and the pencil falls gradually to P_4, then to P_5, and so on to P_{10}, where the steam is blown into the air, and the spring being no longer compressed, the pencil falls to the line called the atmospheric line, a line which will again be spoken of presently.

At P_{11} the engine begins the return stroke, and up to P_{19} the steam continues to escape into the air; at this point the valve closes, and what is left in the cylinder is compressed until the point P_{21} is reached, when steam is admitted again and the spring

compressed up to P_1. From this curved figure we must now find what power the engine was exerting. Let us first consider the area between the two lines A B; during this first tenth of the stroke the pencil of the indicator shows the full initial boiler pressure from P_1 to P_2; therefore, while the piston of the engine travelled the first tenth of its stroke, it was pressed by steam of full boiler pressure, and the foot-pounds of work it did might be expressed by the initial pressure which is measured from the atmospheric line up to P_1, viz. 45 lbs. on the square inch multiplied by the area of the piston in square inches, multiplied by the distance in feet through which it travelled. Supposing the cylinder to be 10 inches diameter, and the stroke 2 feet $= \frac{45 \times 78 \cdot 5 \times 2}{10}$. Similarly between B and C. But when we come to the space between C D, the pressure is falling from P_3 to P_4, and we must take an average pressure, judging it by the eye, which will be measured up the centre of the space C D, from the atmospheric line up to d; and, so on, mean pressures are taken between each of the spaces D E, E F, &c., to the end of the stroke. On the return stroke the pencil runs for some distance along the atmospheric line, and when we get back to P_{19} it rises rapidly. Now when the pencil rises on the return stroke it is because there is some vapour in the cylinder which has not got away quickly enough, and which we are compressing, so that from the foot-pounds lifted by the engine, on the travel from A to K, we must now deduct the foot-pounds it has lifted in compressing steam on the return stroke, and this amount is found in precisely the same manner as the first, by measuring from the atmospheric line up to the bottom line of the diagram. The result is the total foot-pounds done by this end of the cylinder during one complete revolution, and the same process gone through with the diagram for the other end of the cylinder gives the foot-pounds it has lifted. In practice, however, instead of measuring all the pressures on the stroke from A to K up to the top line of the card from the atmospheric line, as we have done above, and then measuring from the atmospheric line up to the bottom line of the card to get the back pressures, and deducting one from the other, it is quite clear we may at once measure the distance $a\,b$, and consider that as the effective pressure, and that instead of multiplying each

of these by the distance in feet and by the area of the piston, we may add them all together and divide them by 10, which will give us a mean pressure throughout the stroke. This one mean pressure we may multiply by the stroke in feet and by the area of the piston, and similarly for the other end of the cylinder; but we may simplify the calculation still further by adding the mean pressures found for each end of the cylinder together, and dividing it by two, which gives a mean pressure for both ends of the cylinder.

If the engine is a double-acting one, the diagrams for each end of the cylinder are usually taken on the same card, giving a double figure, as in Fig. 2. Each of these diagrams has its own mean pressure, and they are rarely the same. In practice they are nearly always treated as above; the horse-power for each end of the cylinder being rarely calculated separately. In the present instance the mean pressure of the left-hand diagram is 22·25 lbs., and that of the right-hand one 25 lbs.: the mean of both is 23·625. To find the foot-pounds raised per minute by one cylinder we multiply the mean pressure, 23·625, by twice the stroke in feet, by the revolutions per minute, and by the area of the piston in square inches.

James Watt calculated, from experiments which he carried out, that a horse working eight hours a day was capable of raising 33,000 lbs. one foot high per minute. We have seen above how to calculate the number of foot-pounds raised by the engine per minute, and if we divide that number by 33,000 we get the indicated horse-power of the engine.

If the engine is a single-cylinder one, the indicated horse-power is

$$\frac{\text{Area of cylinder} \times \text{Mean pressure} \times \text{Revolutions} \times 2 \times \text{Stroke}}{33000}.$$

If the engine had been a double-cylinder one, the power of both cylinders would have to be added together to get the power of the engine.

Where there are numbers of cards all taken from the same engine to be worked out, a further simplification is made. Instead of multiplying the area of the piston by 2, and by the stroke, and dividing by 33,000 each time for each card, we may find what this sum, which is invariable for each particular engine, is, and multiply

it by the mean pressure and the revolutions. This quantity is called the constant for the engine. If the cylinder had been 10 inches diameter the constant would be $\frac{78 \cdot 5 \times 2 \times 2}{33000} = \cdot 00952$.

Then to find the horse-power we multiply $\cdot 00952 \times R \times mp.$* In the card we have been considering it is

$$\cdot 00952 \times 140 \times 23 \cdot 625 = 31 \cdot 4,$$

supposing the engine to run 140 revolutions a minute.

We have seen that an indicator card is usually run out, that is to say, the mean pressure of the card is usually ascertained, by reading off with the scale the different mean pressures in each of the ten spaces, and then adding them together and dividing them by ten, and this is correct provided each reading is a correct one, but the following is a far better and easier method. A long strip of paper, $\frac{1}{2}$ inch wide (the length varies from 6 to 18 inches, according to the nature of the card), has a starting point marked on the edge, near to one end. The strip of paper is then laid along the first tenth of space, and the length of the mean pressure marked off, not read off, starting from the starting point. It is then laid on the second space, and again the mean pressure is marked off, starting from the point where the first mean pressure ended, and so on to the last of the ten mean pressures. By this means the mean pressures in each of the ten spaces are laid end to end. If we now take a rule, and read off how many inches there are in the whole length, and divide them by ten, we get the number of inches in the mean pressure of the whole card. Consider the left-hand diagram in Fig. 2. Suppose when all the mean pressures for each of the ten spaces when added together measured $6 \cdot 95$ inches, then $\frac{6 \cdot 95}{10}$ is the measure in inches of the true mean pressure, and if the scale was $\frac{1}{32}$ inch, that is, if 1 inch stood for 32 lbs., then $\cdot 695$ inch stands for $\cdot 695 \times 32 = 22 \cdot 25$.

Generally expressed, we multiply the total number of inches read off the strip by the scale, and divide by ten.

This is one of the best and safest, if not the very best, way of finding the mean pressure of a card; it is certainly greatly superior

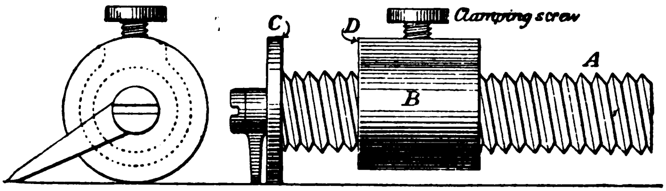
* *Mp.* being used as an abbreviation for mean pressure.

to the method of reading off ten different pressures, and adding them together and dividing by ten, as just described, and it is now very generally followed.

There are one or two other methods.

The author had an instrument made for the purpose similar to the one shown in Fig. 3. The screw A runs freely in the nut B.

FIG. 3.



At the end of the screw a wheel with a milled edge is fixed firmly, and in the centre of the wheel a pointer hangs freely on a stud. The nut is held in the right hand, while the pointer with the wheel is run successively up the centre of each of the ten spaces in the card.

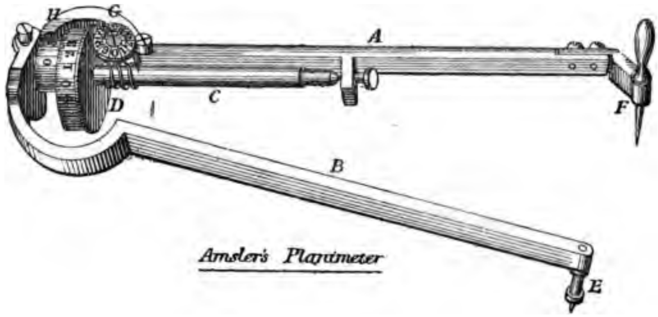
The mean pressure is then read off at once with the proper scale to which the diagram has been taken between the points C D, which will in all cases be the true mean pressure.

The pitch of the screw and the diameter of the milled wheel are so connected that when the wheel and pointer are run 10 inches along a line, the screw will just screw itself out 1 inch; so that the distance C D is always one-tenth of the distance travelled by the rim of the wheel and pointer.

There is a most ingenious instrument called a planimeter, which is now often used for finding mean pressures. Fig. 4 is a drawing of one as made by Messrs. Elliott, of the Strand. The proportions of this one are such that the area of the figure is given in square inches. Thus if we had used it in determining the mean pressure of the left-hand diagram in Fig. 2 we should have read off the number of square inches in the figure; if we divide this by the number of inches between the points A K, we get the number of inches in the mean pressure, and, multiplying by the scale, we get the mean pressure in lbs. In working out a number of cards with a planimeter it is most important to remember that the length of

the card must be taken into account, because this generally varies to a slight extent in cards taken from the same engine with the same indicator, and it does not do to assume a common length for them all.

FIG. 4.



The following are Messrs. Elliott's directions for using the instruments:

"The planimeter, when ready for use, as in the above diagram, rests upon three points D, E, F; these are respectively: 1st, a point of the circumference of the divided wheel D; 2nd, a point of the tracer F at the end of the arm A; 3rd, a point E at the end of the other arm B, which is kept fixed during the time of operation.

"Place the point E at a convenient distance from the figure to be measured, so that the tracer F may traverse the entire periphery of the figure; but if the figure is too large to allow this it can be subdivided by drawing straight lines through it, and the contents of the several parts computed separately and added together. Then place the point of the tracer on any convenient starting point in the periphery. When the instrument is thus adjusted, read off the division on the horizontal disc G, also that on the perpendicular wheel and vernier H. Suppose that the horizontal disc gives 3, and the vertical wheel gives 905, namely, 90 on the wheel and 5 on the vernier, this reading must be put down thus, 3·905. Then carry the tracing point round the figure in the direction of the hands of a watch, and when the whole circuit has been made, observe the reading again. Suppose them 5·763, then subtract the former reading from the latter, the result will be 1·858; multiply

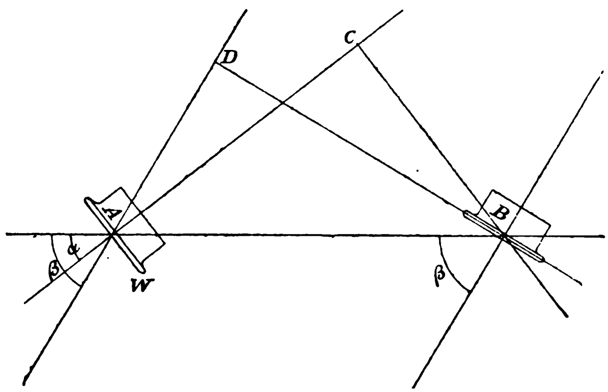
this by 10, and you will get the contents of the figure in square inches, namely 18·58.

"Notice must be taken whether the disc G has made an entire circuit, if so, 10 must be added for every revolution to the whole number; thus, if the disc had gone once round, the second reading would have been 15·763; if twice round, 25·763, and so on."

There will be found in Porter's book on the indicator a description, by Professor Merrifield, of the principles on which this instrument works. I give here another explanation, perhaps more suited to an elementary work.

In Fig. 5 a wheel W is shown, similar to the Planimeter wheel, which rotates freely round the axis A C. Suppose the wheel to be

FIG. 5.

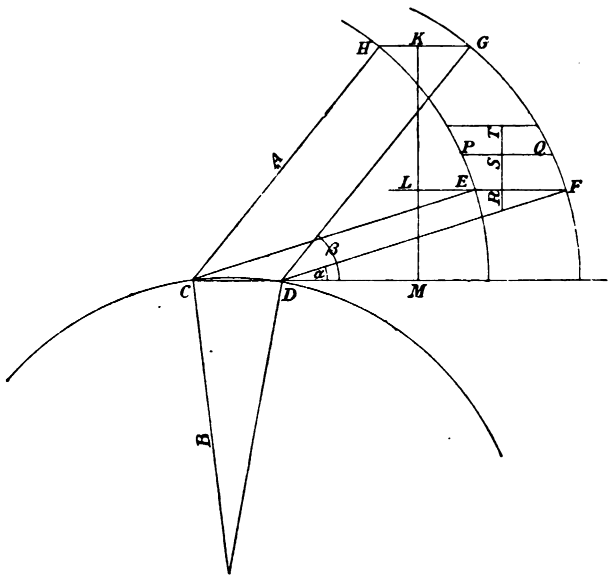


moved along the line A B from A to B, keeping the axis round which it revolves parallel to A C; it is clear that the effect upon the wheel is precisely the same as if, instead of going straight from A to B, it had firstly been moved from A to C, and then from C to B. While travelling from A to C the wheel would not have revolved at all, but while going from C to B the amount of revolution measured along the periphery would have been exactly equal to C B, that is, equal to A B sin. a, where a is the angle of inclination of the axis of the wheel to the line A B. When the wheel is at B change this angle of inclination from a to β, and then move the wheel back to A along the line B A, keeping the inclination of the axis always equal to β; the amount of revolution of the wheel

is now A B sin. β, but the revolution, while going from A to B, is in a contrary direction to the revolution while going from B to A. When the wheel is back at A the total revolution is equal to the difference between the two revolutions, or A B (sin. $\beta -$ sin. a).

Suppose in Fig. 6 that B represents one leg of the planimeter and A the other. Let the leg B describe a circle round the point

Fig. 6.



O, and take two points C and D very near together on the circle, from the points C and D let two other circles be described with the other leg, A, for a radius. Join C D, and produce the line towards M: take any points H and E, and draw through them lines parallel to C D, viz. H G and E F; draw the line K M at right angles to H G from any point K. Now we shall proceed to consider the area of the curved figure E F G H. This is obviously E F $\times$ K L, for if we draw P Q parallel to E F, we know that P Q and E F are both equal to C D, because the circle has just been moved laterally from C to D. P Q = E F, and if P Q be taken near enough to E F, the area of the element P Q F E may be made to differ by as little as we please from E F $\times$ R S, or, in other words, the area of the element above P Q E F = E F $\times$ R S.

Similarly the area of the next element above $PQ = PQ \times ST = EF \times ST$

$$\therefore \text{ the area of the two elements } = EF \times (RS + ST) \\ = EF \times RT;$$

and similarly the area of the figure E, F, G, H = $EF \times LK$.

(Put shortly, the area is $\int EF \cdot h = EF \times LK$), but

$$LK = KM - LM \\ = CH(\sin.\,\beta - \sin.\,\alpha);$$

$$\therefore \text{ area of } EFGH = EF \times CH(\sin.\,\beta - \sin.\,\alpha).$$

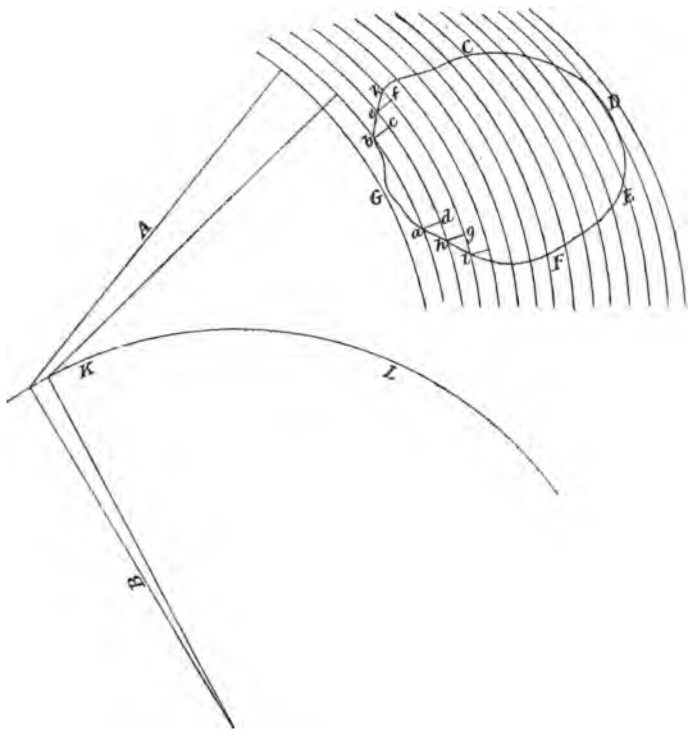
Now if we run the planimeter wheel or the pointer, which is practically the same, round the Fig. E F G H, it is quite clear that the revolution of the wheel while passing from F to G is exactly equal and opposite to the revolution while passing from H to E. These two balance each other and may be neglected; but the revolution while passing from E to F is $EF \sin.\,\alpha$ in one direction, and $EF \sin.\,\beta$ in the opposite direction while passing from G to H. The total revolution while going round the element E F G H is therefore $EF(\sin.\,\beta - \sin.\,\alpha)$.

We have seen that the area of the element is $CH \times EF(\sin.\,\beta - \sin.\,\alpha)$, and the revolution of the wheel is $EF(\sin.\,\beta - \sin.\,\alpha)$, $\therefore$ the area $= CH \times$ revolution of the wheel, and C H is a constant or invariable quantity, therefore the area of the element varies directly with the revolution of the wheel: we have only to note the amount of revolution of the wheel to learn the area.

Now suppose we have to find the area of an irregular shaped figure, such as the Fig. C D E F G, Fig. 7. This figure may be divided into a series of elements by circles struck from the circumference of the circle K L. Now starting from the point a, we go over $a\,b\,c\,d$ and back to a; this gives us the area of $a\,b\,c\,d$. Now to get the area of the next element we go over $a\,d\,c\,e\,f\,g\,h\,d\,a$. This gives the area of the two elements; but we have gone over the line $c\,d$ twice in opposite directions, and over the line $a\,d$ three times, two of which balance, so that to get the area of the two elements we might at once have run over the lines $a\,b\,c\,e\,f\,g\,h\,d\,a$ But the effect of passing over $b\,c$, $c\,e$, is precisely the same as passing over $b\,e$ (if the circles are struck very near together), so

that to get the area of the two elements we may pass directly over *a, b, e, f, g, h, a*. Similarly for the third and fourth, &c., elements; finally, to get the area of the sum of the elements we pass over the

FIG. 7.

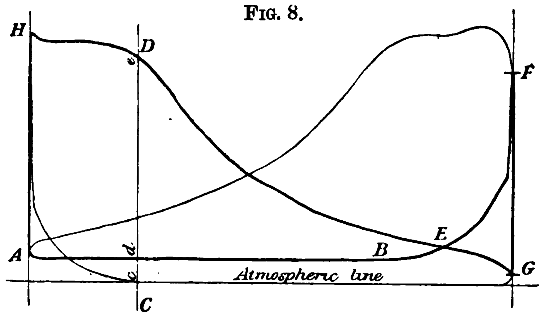


periphery of the figure. The nearer the circles approach to one another the more nearly do the areas of the sum of the elements and of the figure approach to one another, and eventually become equal.

We assumed in Fig. 2, page 6, that the mean pressure of the steam for each end of the cylinder was to be found by adding all the pressures *a b*, *b c*, &c., together. This gives us the total horse-power correctly, but we must not suppose that such pressures as *a b*, *b c*, *c d* ever came on the piston per square inch. This is easily seen if we take a couple of cards from a cylinder from which at one end the exhaust steam cannot escape readily, as in Fig. 8, giving us an exhaust line A B in the right-hand diagram. Now

at any point C D we have a pressure of steam equal to *c e* pushing the piston forward, and a back pressure equal to *c d* pushing it back, so that our effective forward pressure is *d e*. The back pressure is *c d*, because it is pressure on the opposite side of the piston to the steam pressure, and therefore appears in the other card. The power exerted by the engine when making the left-hand diagram is really therefore that which the diagram between the thick black lines would give, and, as beyond the point E, the back pressure actually is greater than the steam pressure, the power developed as shown by the area E F G must be considered negative, and must be deducted from the power given by the figure A H D E B. If both diagrams are worked out in this way, the whole horse-power will be precisely the same as if it had been worked out in the manner already spoken of, which is far simpler; but if we wish to know what the piston is doing on each stroke we must work the cards out as in Fig. 8. Of course the exhaust lines are never in practice as they are drawn in Fig. 8; but they

Fig. 8.



generally do differ slightly, and in the high-pressure cylinder of a compound they do so considerably. For the total horse-power this change of the exhaust lines is of no consequence; but we can now see that this must be taken into account in determining the strains on crank axles if great accuracy is desired.

From what has been said about the indicated horse-power, it will be seen that it depends upon two variable quantities, viz. the number of revolutions per minute and the mean pressure in the cylinder. These two multiplied together, and then the product multiplied by the constant of the engine, give the indicated horse-

power. But there is another term applied to engines to express their power, nominal horse-power, which has no meaning whatever now. As, however, it is frequently used, I will shortly explain how the two terms came into use.

About the year 1784 James Watt was making engines for the London brewers, who were at that time using horses for pumping purposes. When they wished to know what power one of Watt's engines would exert, they asked him how many horses it would be equivalent to?

James Watt, in order to determine this matter, carried out some experiments, and concluded that when such a horse as they used walked at the rate of 2½ miles an hour, it was capable of raising 33,000 lbs. one foot high a minute. He called such an exertion of power one-horse power. Smeaton had made experiments prior to this, and found 22,000 lbs. raised a foot high a minute to be much more nearly the power of an average horse. But Watt's estimate of 33,000 foot-pounds, which is much too high, has been universally adopted, and so long as Watt's estimate is used as a standard of comparison only, and for no other purpose, it is of no consequence whether it is too high or too low.

To determine the horse-power of an engine, Watt and those who immediately followed him supposed every square inch on the piston to be able to lift a weight of 7 lbs.; and when doing this work it was found that the piston would move through from 200 to 256 feet a minute in a double-acting engine. The area of the piston in square inches multiplied by 7 lbs. × the number of feet travelled through per minute, divided by 33,000, was called the horse-power. And it is curious to observe that the 7 lbs. mentioned here were not supposed to be 7 lbs. of mean steam pressure on the piston, but 7 lbs. of pressure actually transmitted through the pump rods, and was equivalent to considerably more than 7 lbs. of steam pressure, as all the friction of the machine had to be added, as well as the power required for the air pumps, &c.

Smeaton considered that in his improved engines of Newcomen's type, which preceded Watt's, that, while his mean steam pressure was 10½ lbs., 1·74 lb. or 16½ per cent. of this was exerted in overcoming friction.* As time elapsed, however, steam

* Farey, 176.

pressure was raised, and instead of our using steam just capable of balancing the atmospheric pressure, we commonly have steam capable of balancing 60 or 70 lbs. per square inch in addition in marine work, and much more than that in some other classes of engines: and further, the piston speed has altered immensely. The old system of estimating the horse-power at 7 lbs. pressure still continued, but as this now differed from the power actually developed, the one was called the nominal horse-power and the other the indicated. At first, during Watt's early time, they were the same, but when they began to differ the old name was retained, and for a certain period of time, as long as engineers worked their engines under similar circumstances of pressure, piston speed, and expansion, it still had a meaning: but now that engines vary so much in all points, some having condensers and others none, steam pressure, and in point of piston speed the term nominal horse-power, has lost all meaning excepting to individual firms, who may use it as a term of comparison. To tell an engineer that an engine is 60 horse nominal is simply to give him an estimate of its power at 7 or 7½ lbs. mean pressure and some arbitrary piston speed, but tells him literally nothing of the developed power, or the quality of the engine.

When Watt first used the term horse-power it meant work actually done in the pumps, and not the work done by the steam. Now it means the work done by the steam: from this the friction of all the moving parts must be deducted before we get at the power transmitted through the shaft. The latter meaning of indicated horse-power is very much the most convenient, as it is almost impossible to know what to deduct for friction, which it would be necessary to do if we calculated the power as Watt did. The better an engine is made the more the power sent through the shaft is, as compared with the power exerted by the steam.

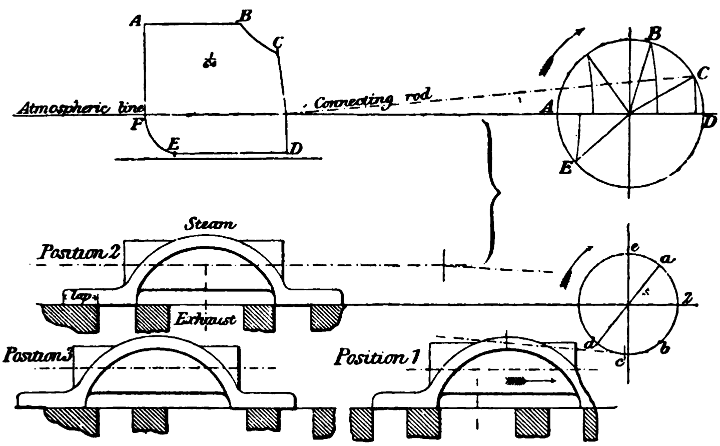
Since no meaning of any value can now be assigned to the term nominal horse-power, I have not thought it necessary to give any rule for ascertaining the nominal horse-power of any engine beyond the old rule of James Watt referred to above.

I shall now take a simple case of a cylinder acting directly like a locomotive and not condensing, and endeavour to show the connection existing between the indicator diagram, the piston, and the slide.

Although one great use of the indicator is to enable us to see how much power the engine is exerting, this is not its only use, for it tells us how the engine is exerting it, and it is to show this that I now proceed.

Take a cylinder 17 inches diameter, 2 feet stroke, with exhaust port $3\frac{1}{4}$ inches wide, steam port $1\frac{1}{2}$ inch, bars $1\frac{3}{8}$ inch, and travel of slide $4\frac{1}{4}$ inches, with a lap of $1\frac{3}{4}$ inch. Fig. 9 shows the slide and slide face drawn to 3-inch scale in three different positions; it also shows an ideal diagram for such a cylinder. The path of the centre of the crank pin is shown to $\frac{3}{8}$-inch scale by the circle A B C D E. The path of the centre of the eccentric is shown by the circle *a b c d e* to $1\frac{1}{2}$-inch scale.

FIG. 9.



I have supposed in this ideal diagram that no steam is admitted until the piston has reached the very end of its stroke, viz. A, and is about to begin the return stroke. This is, or at least should ever be, the case in practice, as will be presently seen.

When the piston is at the end of its stroke, that is at A, it will be found that the eccentric will be at *a*, and the slide as shown in position 1. The instant movement begins, the steam rushes into the cylinder until the pressure in it is equal to the pressure in the boiler; supposing this to be 30 lbs., and a $\frac{1}{64}$th-scale to have been used in the indicator, the pencil would rise $\frac{15}{32}$ inch above the atmospheric line to the point A. The axle

would now begin to revolve in the direction shown by the arrow, carrying the eccentric with it; the eccentric centre would travel from *a* to the point 2, continually making the steam opening larger, and then as the axle continued to revolve the eccentric centre would travel from 2 to *b*, continually diminishing the steam opening, until eventually at the point *b* the steam opening will be closed, the slide again in position 1, but travelling in the opposite direction, and the crank-pin centre at B.

This point B is called the point of cut off. The axle goes on revolving, but no steam can now get into the cylinder; that steam which is already there begins to expand, the pressure becoming less as the volume becomes greater, and this expansion continues until the crank pin reaches the point C, when the slide will be found in position 2. During the expansion the pencil of the indicator falls down the curved line B C as the pressure beneath the indicator piston diminishes. By looking at positions 1 and 2 of the slide it will be seen that the expansion goes on during the time occupied by the slide in travelling a distance equal to the lap. The slide is exactly at half stroke when in position 2.*

The moment the axle revolves further the steam begins to escape from the cylinder into the open air, or into the condenser if it is a condensing engine, more and more rapidly as the opening for its escape enlarges, until the pressure in the cylinder is equal to that of the atmosphere, or in the condenser, as the case may be. This fall in pressure gives the line C D on the diagram. The slide has reached the limit of its travel when it is as shown in position 3; it then returns to the position 2, when the crank pin will be found at E; no more vapour can now get out of the cylinder, and that which is there, being unable to escape, is compressed by the piston, which is now nearing the end of its stroke.

This compression gives the rising curved line E F on the diagram, at which point F we are back at the point from which we originally started.

This compression, like the expansion, lasts while the slide travels through a distance equal to its lap.

The outside lap of a slide is the distance which the face of the slide overlaps the steam port on the steam side, when the slide is

* The effect of the short eccentric rod is for the present neglected.

at half stroke, as in position 2. The inside lap is the amount which it overlaps the steam port on the exhaust side; in this case there is none.

The diagram A B C D E F is then the diagram which an indicator with $\frac{1}{64}$ spring would give, supposing the engine to work slowly so as to enable the steam to get into and out of the cylinder rapidly, and that no steam is admitted until the piston is just about to alter the direction of its movement, that is, the very end of its stroke.

The diagram shows that while the piston travelled from A to B we had full boiler pressure in the cylinder.

This measures $\frac{1}{2}$ of an inch approximately, and the length of the whole card measured in the direction of the movement of the piston is $\frac{3}{4}$, so that we are cutting off steam at $\frac{2}{3}$ of the stroke. At the point C the exhaust begins, the distance A C is about $\frac{23}{24}$ of the whole length of the card, so we see that the exhaust in this particular instance does not take place until the piston has very nearly reached the end of its stroke. Now, measuring backwards from D, we find the distance to E to be about $\frac{21}{24}$ of the whole stroke. At this point the exhaust is closed, and cushioning begins and continues until the steam is again admitted from the boiler.

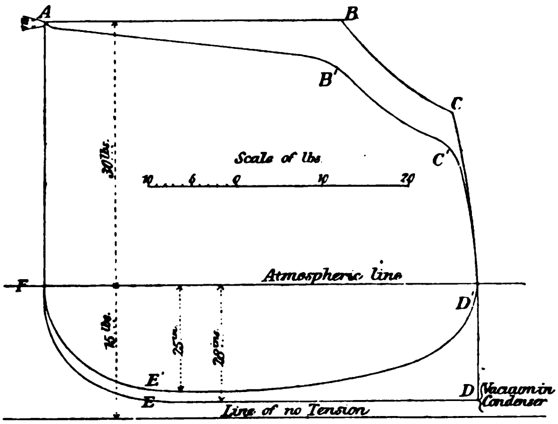
The diagram then enables us to see what power the engine is doing by giving us the means of finding the mean pressure, and further tells us at what point the steam was cut off, where the expansion ended or the exhaust began, and where cushioning began, and many other things which will be referred to later on.

Such a diagram is never produced in practice, for engines never run slowly enough to give such a one. On Fig. 10 is given the same diagram, but to three times the size, the letters A B C D being retained to show the same points. Inside it is drawn what would, probably, be the actual diagram obtained from the engine itself. I shall start from the point A, and work round the card, giving the reasons for the difference between the two, for it is important that the relationship between them should be properly understood.

Firstly, then, at A we seldom find the same pressure in the cylinder which the gauge on the boiler shows, the amount of this difference varies through considerable limits; in this particular case I have taken it as 1 lb., which is small. The larger and shorter

the steam pipe is made, and the larger the port passages are made, the better the jacketing, and the slower the engine goes the less this difference becomes; but if the cylinder is much cooled at each exhaust the difference will be increased.

Fig. 10.



Turning to Fig. 9 we observe that if the crank axle revolves at a uniform speed, which it may be said to do practically, that it may revolve from the point A through a considerable number of degrees, while the piston moves forward very slightly; on the other hand, looking at the diagram showing the path of the eccentric, we see that at the point a, corresponding to A, the eccentric is in such a position that a few degrees revolution of the axle will produce a considerable movement of the slide; so that, starting from the point A, the piston is moving slowly, while the steam opening is increasing rapidly. A little later on, at half-way between A and B, the piston has begun to move quickly, while the slide has reached the point where it gives the maximum steam opening, and has begun to diminish that opening by travelling backwards: now the piston is moving more and more rapidly, while the slide is actually closing the steam opening.

And when the crank pin has got round to B the piston is moving almost as rapidly as it ever did, and the steam opening is actually reduced to nothing, in fact the steam is cut off. The effect of this continually accelerated movement of the piston, which necessitates a continually increasing supply of steam at the back of

it to fill up the space in the cylinder, combined with the continually diminishing facilities for the steam getting into the cylinder, is that the steam pressure gradually drops in the cylinder, as shown by the line A B′, doing so most rapidly just before the point of cut off at B is reached. This point I have marked B′ on the card, which I will for the future call the actual card.

While the piston and crank pin move from B to C, Fig. 9, the centre of the eccentric moves from *b* to *c*, and the slide is pushed from position 1 into position 2. During the whole of this time the port in the cylinder was closed. It is kept closed during the time required by the slide valve to travel the length of its lap, and from this it is clear that the amount of expansion in a cylinder is to a certain extent determined by the amount of lap given to the slide valve.

While the piston then travelled from B to C, the steam pressure would gradually fall owing to the expansion of the steam, and a curved line B′ C′ (see Fig. 10) would be formed. This curved line would approximately follow Boyle and Mariotte's law, which says that the pressure varies inversely as the volume, that is, taking a particular instance, if you double the volume you halve the pressure. The curved line B′ C′ is that part of the card which is formed during the time the steam is expanding, and the longer this line is, within limits, the more economically the steam is being used. The instant the axle revolves beyond the point C (see Fig. 9), the eccentric centre moves towards *d*, and the port passage in the cylinder is placed in communication with the passage to the condenser; the steam begins to rush out, and the steam pressure in the cylinder to fall very rapidly. Here again, by looking at Fig. 9, we see that about the point D the piston is moving very slowly while the slide moves rapidly, and rapidly enlarges the opening from the cylinder to the condenser, so that the steam has plenty of time to escape, and the curved line C′ D′ is formed (see Fig. 10). After the crank pin is over the centre, and has begun to return towards E (Fig. 10), the steam which was not at out of the cylinder still rushes out, and the curved line D′ E′ is formed; at the point E′ the slide is back in position 2, shown in Fig. 9, and no more steam can escape from the cylinder. The vacuum in the cylinder is never so good as that in the condenser, the difference

between them varying very much; the line D′ E′ is therefore above the line D E, Fig. 10. If we suppose a vacuum of 28 inches mercury, as shown by the line D E, to be in the condenser, we should probably get a vacuum of 25 inches in the cylinder at the best place, as shown by the line D′ E′. And we may observe in passing, that supposing the barometer had stood at 30·75 inches, and that we had had a perfect vacuum in the cylinder, the line D′ E′ would have fallen down to the place marked line of no tension or no vapour pressure.

As it is, we have some vapour left in the cylinder, and the instant the axle revolves from the point E the cylinder port is closed by the slide face, and remains so during the time occupied by the slide in travelling over its lap, while at the same time the piston approaches the end of its stroke and compresses the vapour, giving the curved line E′ F′; at this point F′ the slide opens a passage again for the admission of steam, and we begin the card over again.

From this it is clear that the theoretical card is no very perfect guide to what the actual one is, but nevertheless it is the nearest approximation we can get, and it is consequently our safest guide; but nothing but great experience will enable anyone to say with anything like certainty what the real card, as taken from the engine, will be.

In the preceding diagram we mentioned that during the travel of the piston from E to F the vapour which was left in the cylinder was compressed, and formed the line E′ F′.

Although it might at first sight appear that in compressing this vapour the engine was actually destroying its own work, or doing useless work, this is not really the fact to any great extent, for this very same compressed vapour expands as soon as the piston begins to move away from the end of the cylinder on the return stroke, and gives out exactly the same amount of work, minus the friction, in expanding, which had been put into it while it was being compressed. It acts, in fact, very much like a spring would do, and so far from this compression being a waste of power, it serves a most useful purpose. If it were not for it the piston, piston rod, and connecting rod, which are heavy bodies moving rapidly towards the end of the cylinder, would by virtue of their

momentum put a heavy strain on the rubbing surfaces of the connecting-rod ends and axle bearings, and on the holding-down bolts. As it is, the compression of the vapour requires a large amount of power, and this power is the momentum of the moving parts mentioned. By compressing, then, we not only save friction on the rubbing surfaces, but the heavy jars are prevented which would come on all the joints at the moment when the motion of the piston was reversed: and this compression, if not carried too far, is economical in another respect.

If absolutely no steam or vapour were left in the cylinder at the end of the stroke, the space to be filled by the incoming steam would be the port passages + the clearance between the piston and the end of the cylinder, and then, as the piston moved forward, the cylinder would have to be filled with steam up to the point of cut off, but if by compression the port passages and clearance were already partly or wholly full of steam, that amount of steam less would be required at each end of the stroke; and the power lost in friction in compressing it would be small in comparison to the steam saved.

Further, in order to prevent these jars at the ends of the stroke, steam is admitted into the cylinder before the piston reaches the end of the stroke. This is termed giving the slide lead, and partly produces the effect termed cushioning.

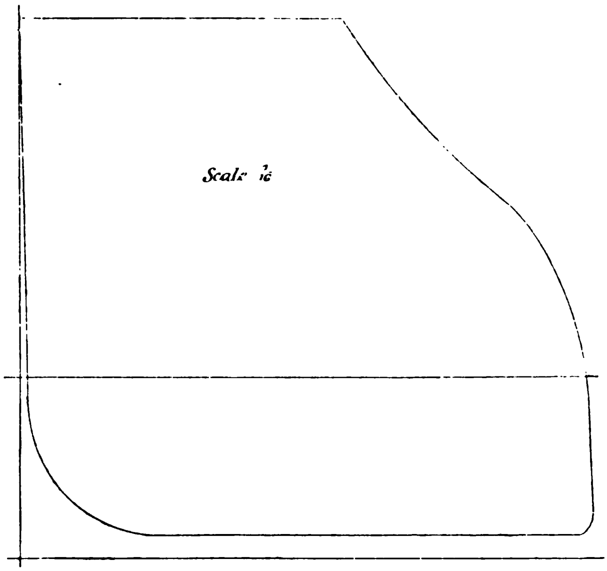
Fig. 11 shows how, by giving lead to a slide, the diagram is altered. Fig. 11 is from precisely the same engine as diagram, Fig. 10, with the exception that the eccentric has been moved round the shaft a little more in advance of the crank than it was in Fig. 10.

One effect of moving the eccentric round is to cut off earlier. Advancing the eccentric round the shaft causes the eccentric to put the slide into all its various positions of admission, cut off, exhaust, cushioning, &c., a little earlier than formerly. We see that in Fig. 10 the steam is cut off at $\frac{6 \cdot 7}{10}$ of the stroke roughly; while in Fig. 11, where lead was given, the steam was cut off at roughly $\frac{5 \cdot 62}{10}$.

It is usual in taking diagrams from an engine to take two on

one piece of paper, both the diagram for the cover end of the cylinder and that for the axle end. We have hitherto only drawn the card for the cover end.

FIG. 11.



It will be easily seen presently that if the connecting rod were of infinite length, and the same lead were given at both ends of the cylinder, then the cards for both the cover and axle ends would be the same; but in practice these conditions are never met with except in very small engines, and consequently the diagrams are not the same.

In Figs. 12 and 13 I have kept the lead at both ends of the cylinder the same, but varied the length of the connecting rod; in Fig. 12 the connecting rod is supposed to be four times the length of the stroke, in Fig. 13 only one and a half times. The lead is one-quarter inch at both ends. These figures show clearly that as the connecting rod grows shorter in proportion to the stroke of the engine the card at the cover end becomes fuller and fuller, as compared with the card from the axle end. In other words, the cut off takes place later as the connecting rod becomes shorter, on the cover side. This is supposing that the same lead is given at both ends of the cylinders, but practically this is varied to a slight

extent; as, for instance, in inverted engines more lead is given on the bottom side of the piston to prevent, by greater cushioning, the effects which would be produced by the heavy falling weights.

FIG. 12.

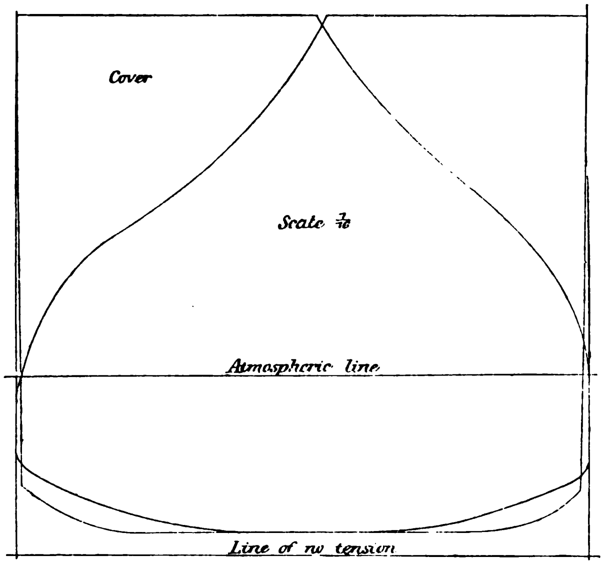
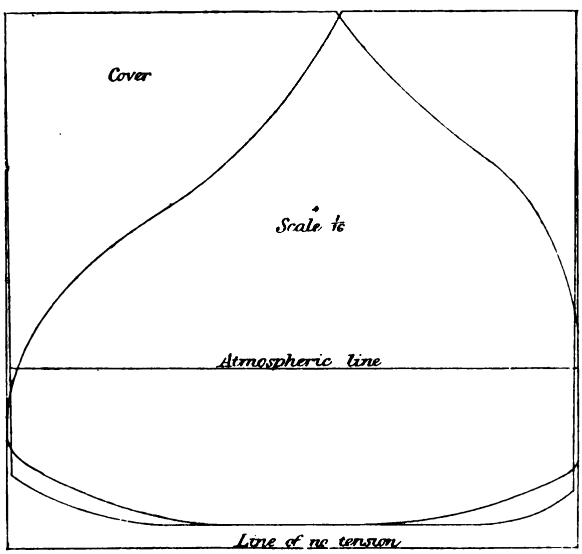


FIG. 13.

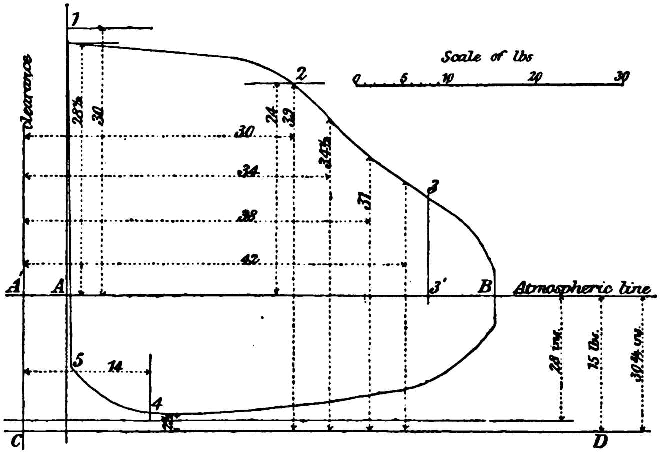


I now proceed to give the system of constructing a theoretical card for an engine. The use of such a card is to determine approximately what horse-power an engine of a certain size will indicate, or to determine what size an engine must be to indicate a certain horse-power. Without some such guide as this it would be next to impossible to determine even approximately the sizes of cylinders required to do a certain amount of work under such a variety of determining circumstances as marine engines are placed in.

I will firstly draw the theoretical card for the engine we have just been considering, with a connecting rod four times the length of the stroke, with 30 lbs. steam.

The line A B, Fig. 14, is drawn to any scale to represent the length of stroke. The distance A A′ is then laid off along the same line to represent what is called the clearance.

FIG. 14.



These clearances are the cubical contents of the port passages, and of the clearance between the back of the piston and the end of the cylinder, or the cover, and supposing them to be, when added together, $\frac{1}{15}$ of the whole cubical contents swept through by the piston, then the distance A A′ would be made $\frac{1}{15}$ of A B. In the present example $\frac{1}{10}$ has been taken, so that the line A A′ is $\frac{1}{10}$ of the length of the line A B. It is necessary to take these cubical contents into account, for the passages and clearances are filled

with steam at each stroke, and expand just precisely the same as the rest of the steam in the cylinder does after the steam has been cut off, and the vapour left in them is compressed in the same way.

Starting then from the point 1 on diagram we shall probably have, instead of 30 lbs. pressure, only $28\frac{1}{2}$ lbs., and this pressure will be further diminished until at 2, the point of cut off, we should, probably, not find more than 24 lbs. of steam. The reason why the steam line would be curved in the manner shown has already been spoken of. This 24 lbs. is measured from the atmospheric line, and the meaning is that if you had a hole in the side of the cylinder exactly of the area of 1 square inch with a valve in it, you would have to put a weight of 24 lbs. on to the valve to just balance the steam pressure from the inside.

We will now see how to determine the curve of expansion. I have supposed that the barometer stands at $30\frac{3}{4}$ inches of mercury, which represents a pressure of about 15 lbs. Measuring off 15 lbs. with our scale (the same one used for the steam pressure), below the atmospheric line, we get the line which an indicator, pressed by the atmosphere on the top side of the piston, to the extent of 15 lbs. to the square inch, and in connection with a perfectly empty or vacuous space on the bottom side, would give: This is the line C D. If we now measure up from the line C D to the point 2 we find that we have a total pressure of the steam or vapour amounting to about 39 lbs. In considering all questions of expansion and compression of gases it is this total pressure, dating from the line of no pressure, which we must consider. The atmospheric line, it must not be forgotten, is merely a line showing what the pressure of the atmosphere happens to be at the time, and the expansion curve has nothing whatever to do with it. It is most important that the truth of this remark should be fully realized.

We have, then, a pressure of 39 lbs., and measuring back to the clearance line we find the distance from the point 2 is $1\frac{3}{8}$ of an inch; using the pressure scale for convenience as the measure, we may say that, at a distance 30, we have a pressure of 39.

We have virtually a cylinder with a length of 30 filled with steam at a pressure of 39 lbs.

What will be the pressure at a distance 34 ?

We know that, according to Boyle and Mariotte's law, that the units of pressure of a gas multiplied by the units of volume it occupies, give always a constant quantity (see page 102). Now we have in this case a volume represented by 30 units, and a pressure represented by 39 units; these multiplied together give 1170, which is the constant quantity, and if 34 be multiplied by $34\frac{1}{2}$ approximately, we shall get 1170.

$$34 \times x = 1170;$$
$$\therefore x = \frac{1170}{34} = 34\tfrac{1}{2} \text{ approximately.}$$

Again, at a distance of 38, we say $38 \times x = 1170$; and, therefore, $x = \dfrac{1170}{38} = 31$ nearly, and so we go on until we come to the line 3, 3′, and there the expansion curve stops, for we know that when the piston has travelled from A to 3′, the slide opens a passage either to the condenser, or to the open air, as the case may be. We shall presently see that the statement that steam expands according to Boyle and Mariotte's law, is not strictly true, but the law does give a sufficiently near approximation for all practical purposes.

From the point 3 to the point 4 it is not possible to determine the shape of the diagram with any degree of accuracy. We have not sufficient data for determining at what rate a vapour like steam would escape through a continually varying aperture, out of a chamber of a continually varying size, but we know that the steam pressure must fall in something like the manner shown until we get back to the point 4, where the slide closes the port.

The port is kept closed until the piston has reached the point 5, at which point steam is again admitted into the cylinders. Between the points 4 and 5 the steam is compressed in the same way that it expanded, following to a certain extent the law of Boyle and Mariotte.

There is a point I may draw attention to now, viz. that any practically attainable speed at which the piston moves does not alter the expansion curve directly, but does so indirectly in this way: the faster the engine goes round the less the time allowed to the steam to get in and out of the cylinder, and, consequently, the steam line, instead of starting at $28\frac{1}{2}$ lbs. starts lower, and the

vacuum line is altered by the steam not being able to get out as quickly, and we get the exhaust or vacuum line raised nearer to the atmospheric line. To obviate this the ports are usually made larger in engines running at high piston speeds.

The theoretical card of one end of the cylinder only has been given, but in determining the power of the engine the other one should be made also in precisely the same way. The mean pressure of this card is 31·2 lbs. If this were a single-cylinder engine making 120 revolutions, stroke 27 inches, and diameter 24 inches, the indicated horse-power for one side of the piston would be

$$\frac{(24)^2 \times \cdot 785 \times \frac{27}{12} \times 120 \times 31 \cdot 2}{33000} = 115.$$

To sum this up, we see that to ascertain the indicated horse-power of a proposed engine the first thing to do is to fix the slide gear, the ports, the length of the connecting rod, the stroke and diameter of the cylinder, and then find out where the cut off takes place, and then lay out the rest of the theoretical or ideal card, from which the mean pressure and the indicated horse-power can be found.

The manner in which these parts are determined is explained in various works on the slide valve: but the actual proportions are different with different engineers.

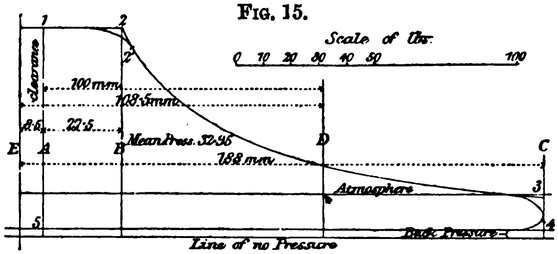
We have hitherto considered the ideal card of an ordinary expansive engine, and we shall now consider the method of determining the probable indicated horse-power of any compound engine.

So far as the author is aware the following method has not been used before, and he has, therefore, been careful to verify it in a large number of instances, where the expansion was from 5 to 9 times, before recommending it. The results are, in most cases, nearer than those obtained in the ordinary manner. The usual method of doing this is to find the volume occupied by the steam at the instant of cut off, and the volume of the low-pressure cylinder, and by dividing the latter one by the former the amount of expansion is determined. A theoretical card is then constructed in which the volume of steam at the instant of cut off is allowed to expand to the extent as found above, and a mean pressure found, from which the horse-power is calculated as if the expansion had

taken place in a cylinder of the diameter of the high-pressure cylinder, but with a longer stroke. Thus if the cut off in the high-pressure cylinder had been at $2\frac{3}{4}$ tenths, and the low-pressure piston had been three times the area of the high-pressure cylinder, both having the same stroke, then the expansion would have been (if we neglect clearances) $\frac{10}{2\cdot 75} \times 3$, or nearly 11 times. Then laying out the card for an expansion of 11 times we get a certain mean pressure, and a virtual stroke of $\left(\frac{2\cdot 75}{10} \times \text{stroke actual}\right)^* \times 11$ $= 3$ times the stroke actual, from which the indicated horse-power would have been calculated as if it had been a single-cylinder engine.

The ideal diagram for a compound engine with cranks at right angles cannot be set out from the slide ellipses, as in the preceding case, with any degree of accuracy, and the process involves a very large amount of calculating. The proposed method is the following: Suppose we wish to find out the probable indicated horse-power of an engine, such as we have been considering, and that the clearance amounts to $\frac{1}{12}$th of the cylinder. Lay off, as in Fig. 15, a distance A D, which represents the high-pressure

Fig. 15.



cylinder. In this case A D = 100 millimetres, the millimetre being a very useful scale for this purpose.† To the same scale lay off A B to represent $2\frac{3}{4}$ tenths of A D = $27\frac{1}{2}$ mm., and to the

* $\frac{2\cdot 75}{10} \times$ stroke actual, is the measure of that portion of the stroke which takes place BEFORE expansion; this multiplied by 11 gives the length after the expansion.

† The diagram was reduced in the press, and, although correct to scale given, is not now in millimetres.

left of A lay off a distance $AE = \frac{1}{12}$th of AD for clearance. When the piston in the high-pressure cylinder has reached the limit of its stroke, the steam will occupy a volume proportional to ED, and at the moment of cut off this steam occupied a volume proportional to EB, therefore the expansion in the high-pressure cylinder is $\frac{ED}{EB} = \frac{108 \cdot 5}{36}$ mm. If the low-pressure cylinder be treated in the same way we shall find the volume occupied by the steam at the end of the stroke is to the volume occupied at the moment of cut off as $108 \cdot 5$ to $62 \cdot 25$, allowance of $\frac{1}{12}$ for clearance to be made exactly as in the first case, supposing it to cut off at $5\frac{3}{8}$ tenths. What I shall in future term the coefficient of expansion must now be found by multiplying the ratio of expansion in the high-pressure cylinder by that in the low-pressure cylinder, viz. $\frac{108 \cdot 5 \times 108 \cdot 5}{36 \times 62 \cdot 25} = 5 \cdot 22$. Now returning to the diagram we must lay off EC, so that $EC = 5 \cdot 22$ times $EB = 36 \times 5 \cdot 22$ $= 188$ millimetres. And now, according to Boyle's law, construct the curve of expansion starting from the point of cut off, and remember that it is from E, viz. the clearance, that the volumes must be measured. Next lay off whatever is considered fair for the back pressure, in this case 3 lbs., and then determine in the usual manner the mean pressure of the figure 1 2 3 4 5. It will be found about $32 \cdot 95$ lbs. Now we must see what is our virtual stroke. The stroke of the engine is 3 feet; the distance AB was taken to represent $2 \cdot 75$ tenths of 3 feet, which is equal to $27 \cdot 5$ millimetres, AC is equal to $179 \cdot 5$ millimetres, and therefore AC represents in feet

$$\frac{179 \cdot 5}{27 \cdot 5} \times \left(\frac{2 \cdot 75}{10} \times 3\right) = 5 \cdot 37 \text{ feet} = \text{virtual stroke.}$$

Instead of considering the engine as consisting of two separate cylinders, we may consider it to be a single cylinder of the diameter of the high-pressure cylinder, with a stroke of $5 \cdot 37$ feet. Supposing the high-pressure cylinder to be 52 inches diameter, then in order to determine the power of the engine going 67 revolutions we say twice the stroke $= 5 \cdot 37 \times 2$, multiplied by the area of the cylinder $52^2 \times \cdot 785 = 2120$ square inches $\times$ the

mean pressure 32·95 × revolutions = 67 divided by 33000 = indicated horse-power.

$$\frac{5\cdot37 \times 2 \times 2120 \times 32\cdot95 \times 67}{33000} = 1521 \text{ I.H.P.}$$

this engine on trial cutting off as above, indicated 1530. It may seem a little curious in this method of estimating the probable horse-power, that the diameter of the low-pressure cylinder, the cylinder which usually does more than half of the work, should be entirely left out of the calculation. The fact is, that both this system and the usual one are quite empirical, and if applied to extreme cases might both of them give wrong results. They merely give us an approximation to the power, and the one that gives it the nearest is the best. In this latter system we only take into account the ratio of expansion in the low-pressure cylinder, and neglect its diameter; in the former system an assumption is made that the steam is doing its full amount of work all the time it is expanding, which is an incorrect assumption. It is also to be noted that we have made very little allowance for wire drawing at the point of cut off. In the first place, in such an engine cutting off at 2¼ tenths, there is very little; and secondly the cut off took place practically at 2′, so that by producing the curve up to 2, we get the virtual cut off as far as the expansion is concerned.

It is from this point of virtual cut off that our calculations must depart.

The virtual stroke of any compound engine is found as follows.

We know (see Fig. 15) that if we lay off E C to the same scale as A B

since E C = E D × ratio expansion in low-pressure cylinder;

but

E D = stroke of high-pressure cylinder + clearance;

therefore

E C = (stroke of high-pressure cylinder + clearance) × ratio expansion in low-pressure cylinder.

Now deduct A E from both sides, and

E C − E A = virtual stroke;

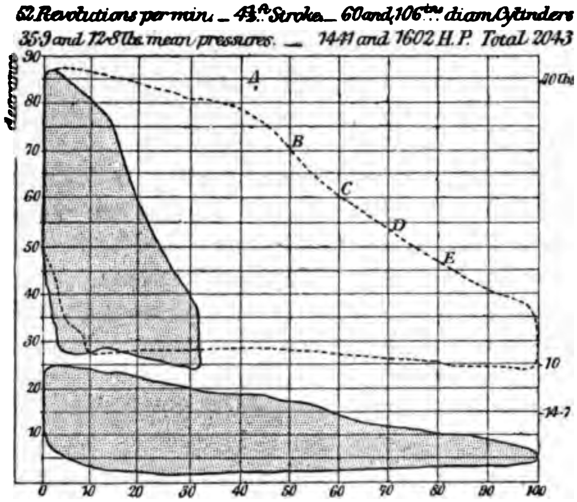
Virtual stroke = (stroke of high-pressure cylinder + clearance) × ratio expansion in low-pressure cylinder − clearance in high-pressure cylinder.

remembering that if the stroke is expressed in feet, the clearance must also be expressed in feet.

I give just one other instance of this method of calculating the indicated horse-power of a compound engine. This is the case of the large engines of the 'Spain,' a vessel belonging to the National Steamship Company. Stroke, 4½ feet. Cylinders, 60 and 106 inches diameter. Indicated horse-power, 3043.

When Mr. F. J. Bramwell read one of the best papers which has been recently written on marine engines, to the members of the Institute of Mechanical Engineers, he showed the cards in Figs. 16 and 17, and stated that the shaded card in Fig. 16 gave

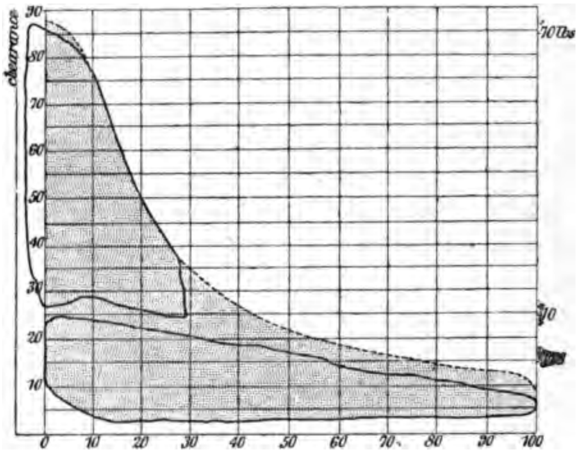
FIG. 16.



10 per cent. less power than the shaded portion in Fig. 17, in which last figure the steam has been supposed to expand continuously in the one cylinder of the same diameter as the low-pressure cylinder of the compound. (In Fig. 16 the high-pressure diagram has been shortened proportionately to the area of the high-pressure to the low-pressure cylinder.) This loss of 10 per cent. is said to take place in the receiver. Now, if the horse-power had been calculated as if the expansion had been the ratio of the expansion in the high-pressure cylinder, multiplied by the ratio of expansion in the low-pressure cylinder, the indicated horse-power,

assuming a back pressure of 3¼ lbs., would have come out very closely 3000 indicated horse-power, or about 1·4 per cent. too low.

FIG. 17.



Figs. 16 and 17 are copied from the Figs. in Mr. Bramwell's paper, published in the 'Proceedings of the Institute of Mechanical Engineers' for 1872. I have to thank the Council of the Institution for their courtesy in allowing me to republish them.

The point A was determined from the points B, C, D, E, which agree very nearly with Boyle's law. The steam has been supposed, in the calculation which gives 3000 indicated horse-power, to continue to expand according to this law.

I may notice, in passing, that if we consider the ratio of expansion to be the volume occupied by the steam when it exhausts, divided by the volume it occupied at the moment of cut off, or A, the ratio is about 6·95. According to the method which gives 3000 indicated horse-power, assuming the low-pressure cylinder to cut off at 5½ tenths, it is 4·3.

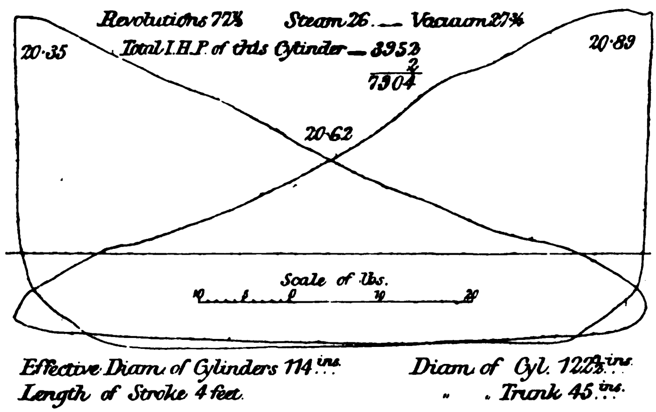
The comparison here made by Mr. Bramwell, between the power developed by equal weights of steam, expanding in a simple cylinder and in a compound engine, is a fair one when understood in a particular sense. If each time that steam was admitted to the cylinders the same amount was condensed by the cooled cylinder in both cases, so that for each pound of steam which left the boiler we should have the same volume and pressure, *after deducting for*

condensation, ready to begin work in each cylinder, then the expansive engine is working at a considerable advantage, as shown by Mr. Bramwell: but if much more of the pound weight of steam leaving the boiler is condensed in the case of the expansive engine than in the compound, then, although the expansive engine may do more work by 10 per cent. with its one cylinder than the compound with its two cylinders, it may be using perhaps 20 or 30 per cent. more steam by weight; that of course means that it is burning much more coal for a horse-power.

That an expansive engine does condense a larger proportion of the steam between the point of cut off and admission and during the expansion than a compound does, is the case in the author's opinion.

In Messrs. Loring and Emery's report of a trial conducted by them of two engines (see page 128), one a compound indicating 266 horse-power, the other an expansive engine indicating 219, the steam pressure being in both roughly 68 lbs., they found that the water condensed in the compound high-pressure cylinder was 6·8 per cent., in the compound lower-pressure cylinder 26·5 per cent., and in the expansive engine 31·7 per cent.; and the consumption of coal was accordingly found to be less in the compound, or in the ratio of 2·43 to 3·13.

FIG. 18.



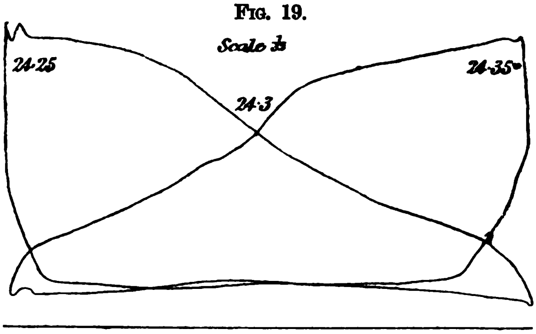
I cannot, therefore, agree with Mr. Bramwell in his remark, that "the reflection is forced on one, that in using steam in compound-cylinder engines, the whole of the high-pressure engine

is absolutely useless as a source of power, and in that respect, therefore, is all waste of weight, of space, and of money." When we speak of power, we must not consider the cost of coal at which it is attained.

There are many precautions to be observed in taking indicator cards, which are only to be learnt by practice, and it is only by a considerable amount of it that an engineer can read a card correctly at sight.

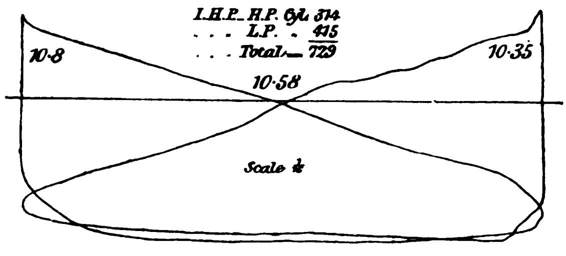
I give a few examples of cards taken from various engines. Fig. 18 is a card from a very large expansive trunk engine, by Messrs. John Penn and Sons, to whom I am indebted for it, and for several others which they have kindly given me. The first thing which strikes one here is the tremendous piston speed, 620

Fig. 19.



Revolutions 94 — Steam 60 — Vacuum 26

Fig. 20.

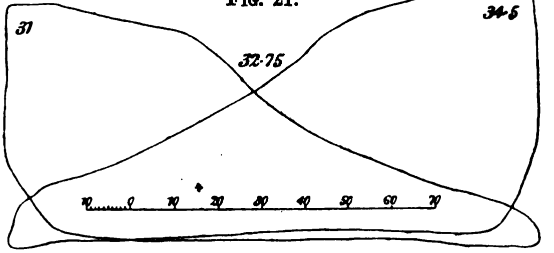


Effective Diam. of H.P. Cy. 38 in. Diam.
" " L.P. . 66 in. Diam.
Length of Stroke 2 ft. 0 in.

feet; this gives rise to considerable wire drawing towards the cut off. The steam pressure, read from the card, shows no drop below that on the gauge, which was, however, no doubt in the engine

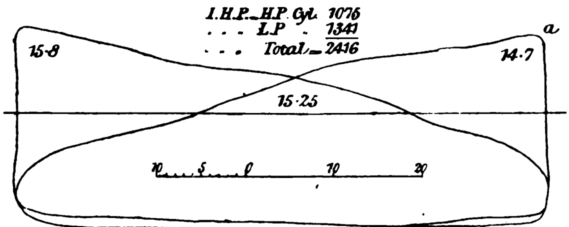
room. The vacuum in the condenser is very good at 27¾, but at times it ran up above this on the trial, and the steam gets out of the cylinders quickly, giving exhaust lines nearly parallel, which is a very important point in a card. It is difficult to form any conception of the force required to stop such enormous masses as the piston and the connecting rod of a cylinder 122½ inches diameter, moving at a velocity of 973 feet a minute, or 16·2 feet a second, in less than one-fifth of a second, and again in the same interval of time to return to them their motion in the opposite direction.

FIG. 21.



Revolutions 57 — Steam 69 — Vacuum 29

FIG. 22.



Diam. H.P. Cyl. 55 ins. Diam. L.P. Cyl. 90 ins. Stroke 4 ft. 0 in.

The next examples are Figs. 19 and 20, taken from a compound trunk engine by Messrs. John Penn and Sons; the steam here shows considerably below the boiler pressure, but the cards show very little wire drawing, and the exhaust is a very fine one on the low-pressure diagram. These engines were highly efficient as regards the consumption of coal compared with the horse-power.

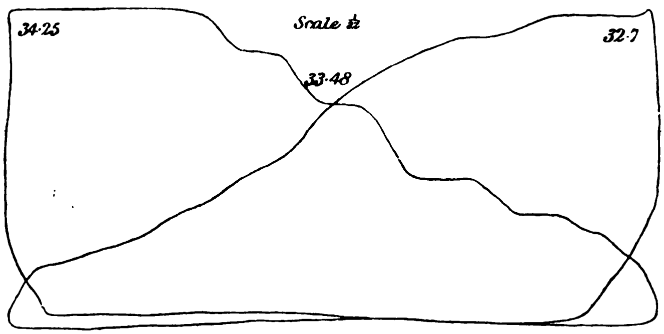
The diagrams, Figs. 21 and 22, are from a compound inverted engine for the National Steamship Company's steamship 'Helvetia,'

by Messrs. J. Penn and Sons. It would be difficult to point out any way in which these cards could be improved. They show practically no wire drawing towards the cut off with a splendid vacuum and free exhaust, and a high piston speed for this type of engine, viz. 456. The sharp corners at *a* and *a*, especially on the low-pressure diagram, show that the steam was dry, one of the best indications of wet steam or very slight priming being the rounding at these corners. The steam pressure was, as will be observed, very high.

The high-pressure slide face was a double-ported one with cut-off slide in the back of the main slide; the low-pressure slide face was treble ported.

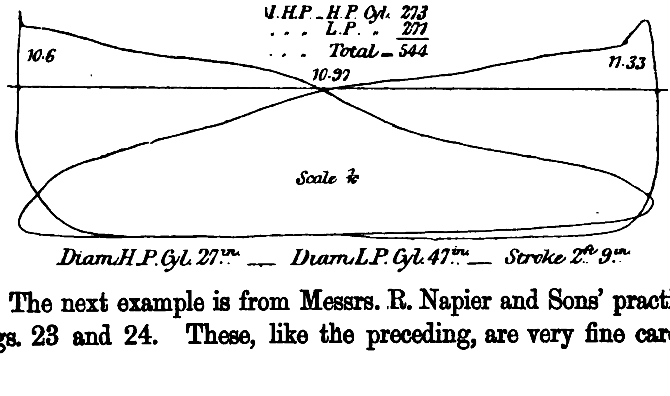
With such cards before us, obtainable with the old, simple, well-tried link motion, it seems hardly wise to be hunting after more perfect diagrams with a complicated gear.

FIG. 23.



Revolutions 85½ Vacuum 26¾

FIG. 24.



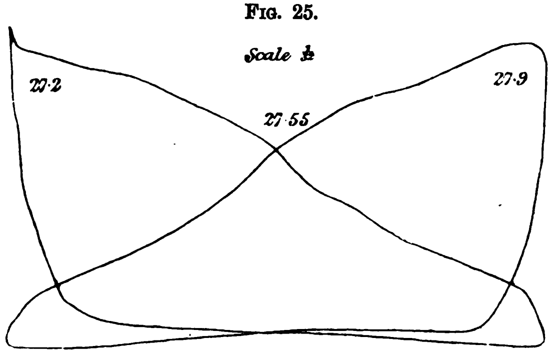
Diam. H.P. Cyl. 27 in. — Diam. L.P. Cyl. 47 in. — Stroke 2 ft. 9 in.

The next example is from Messrs. R. Napier and Sons' practice. Figs. 23 and 24. These, like the preceding, are very fine cards:

the piston speed, 470 feet, is high for such a small engine. Of wire drawing there is none; the wavy expansion line is perhaps due to the indicator. The exhaust lines in the low-pressure cylinder are very good.

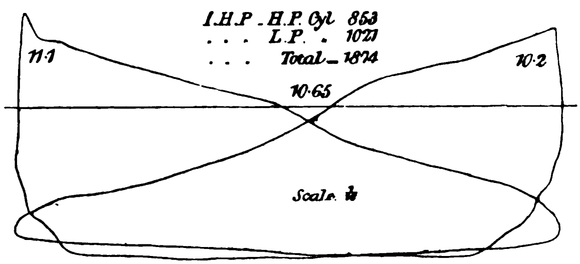
For the diagrams, Figs. 25 and 26, 27 and 28, I am also indebted to Messrs. R. Napier and Sons. They are all taken from the large compound inverted engines of the steamship 'Scholten.'

FIG. 25.



Revolutions 65 — Steam 63lbs — Vacuum 27

FIG. 26.



Diam. H.P Cyl. 50in. — Diam. L.P Cyl. 88in. — Stroke 4ft. 0in.

The exceedingly high piston speed in Figs. 25 and 26 will be noticed at once, 520 feet. The steam line drops a little more than in the case of the 'Helvetia,' but there the piston speed was 456, or 64 feet less. It will be observed also that to meet this high speed, a little more cushioning has been allowed. The all-important question of the vacuum is here very satisfactorily answered. Figs. 27 and 28 are from the same engine, while working at much lower powers: they show an arrangement of slide gear which could not

easily be surpassed so far as the cards show. Messrs. Napier informed me that the high-pressure slide is double ported with gridiron expansion on the back, and the low-pressure slide is double ported with a fixed cut-off plate on the back, an ingenious arrangement by which a practically quadruple-ported slide is obtained.

Fig. 27.

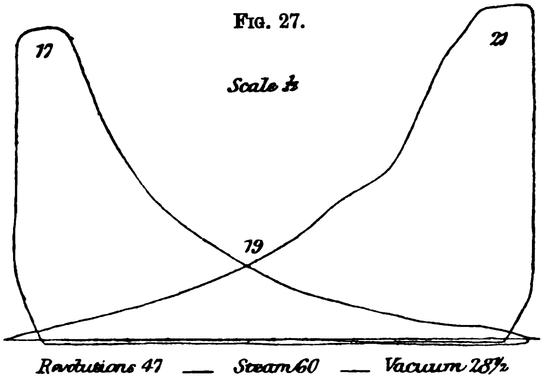
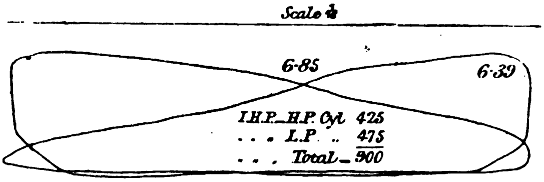


Fig. 28.



Mr. Brotherhood has kindly sent me specimens of cards taken from his patent three-cylinder, single-acting engine. They are shown in Figs. 29 and 30. The exhaust is a very sharp one, and the amount of cushioning would be considered very excessive in marine work, but is perhaps necessary in these engines.

The waviness of the expansion line is due to the use of a weak indicator spring, at such high revolutions. The weakness of the spring makes itself felt more conspicuously than one would be inclined at first sight to suppose, for the following reason: the weaker the spring, the more rapidly the steam will compress it, and consequently the greater will be the velocity of the indicator piston in rising; but the momentum of a body is proportional to the square of the velocity at which it is travelling, and it is this

momentum which carries the piston above the point to which the steam pressure alone would compress the spring. When the momentum has been destroyed by the spring, the spring then forces the indicator piston below the point where it and the steam would be in equilibrium, and then it is again forced too high.

FIG. 29.

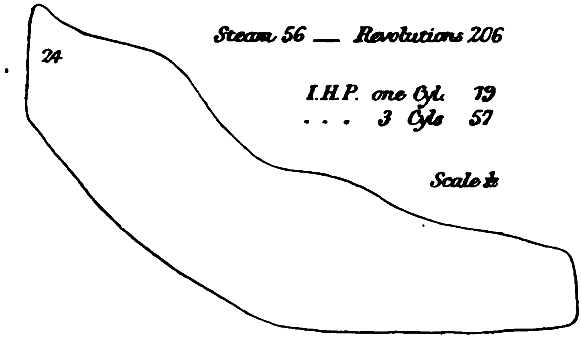
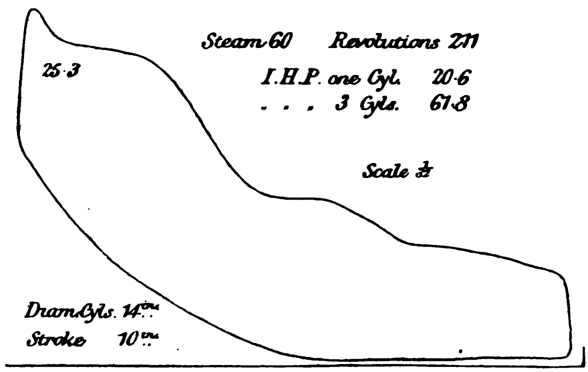


FIG. 30.



These alternate up and down movements produced by the momentum, combined with the lateral movement of the card, give the wavy line. It occasionally happens, as in Fig. 23, that only one of these diagrams shows the wavy line. The cause has not been learnt by the author.

Through Mr. Webb's kindness, I am enabled to give some exceedingly interesting cards taken from locomotive engines of the 'Precursor' class, designed for working the heavy express

passenger traffic of London and North-Western Railway by Mr. Webb. See Figs. 31, 32, 33, 34, 35.

Fig. 35 is taken at a tremendous piston speed, 1180 feet. At the more usual speed of forty-nine miles an hour, the piston speed is 1000 feet.

FIG. 31.

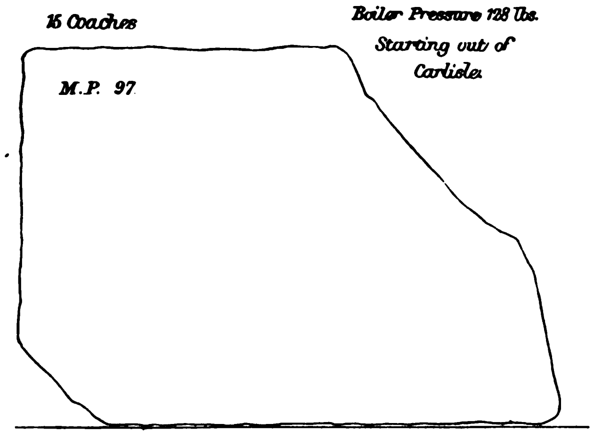
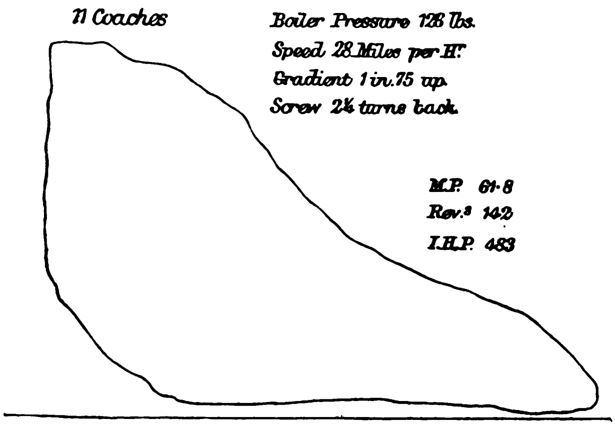


FIG. 32.



It is worthy of notice that between the speeds of 28 and 58 miles an hour, the indicated horse-power varies only from 483 minimum to 592 maximum—a ratio of 1 to 1·22. The gradient and the load dragged did not, however, remain the same. All that

may probably be inferred is that the engine was working near its maximum power all the time. It will be observed also that the steam pressure remained very much the same throughout. The difference between the boiler pressure and the cylinder initial pressure in Figs. 31, 32, and 33 is roughly 8 to 10 lbs.

FIG. 33.

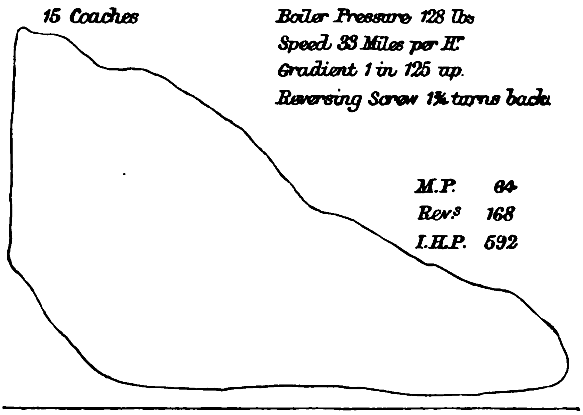
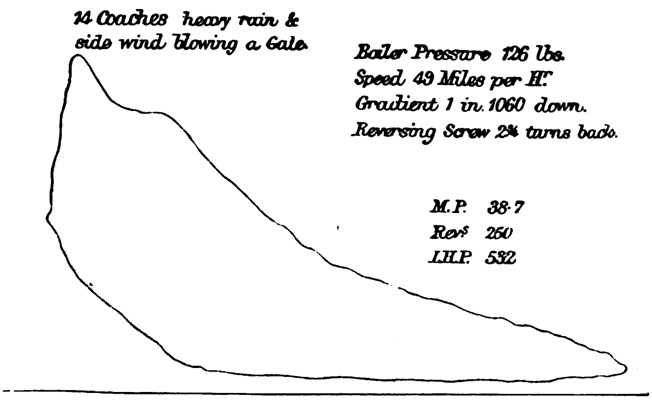


FIG. 34.



For the diagrams shown in Figs. 36 and 37 I am indebted to Mr. G. Salt. They are taken from one of the large stationary engines driving the various machines in the Saltaire Mills.

The piston speed in Fig. 36 is rather high, 720 feet.

The indicator may also be used to show the varying pressures in the air and circulating and feed pumps; but none of these are of any great practical importance.

FIG. 35.

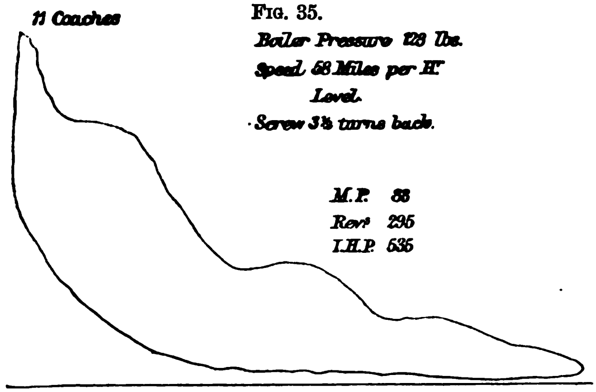


FIG. 36.

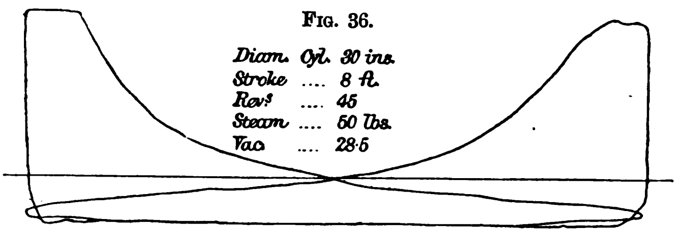
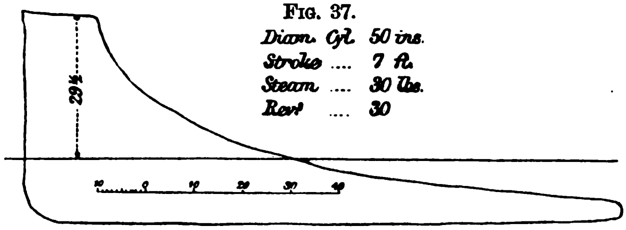


FIG. 37.

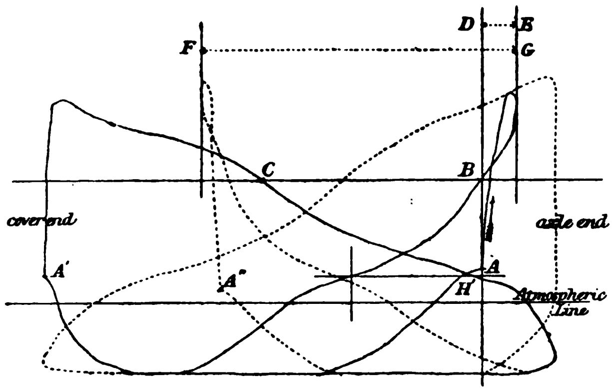


There is another variety of indicator card which is very instructive, an example of which is given in Fig. 38. The author took this from a diagonal paddle engine indicating 98, built by Messrs. J. Penn and Sons. The cylinders, jacketed, were 18 inches diameter and 27 inches stroke; revolutions 52.

It will be seen that in the figure there are firstly the two ordi-

nary cards made by attaching the string of the indicator to some moving part of the engine, the movements of which part correspond with the movements of the piston. The cover end card is drawn in full black lines, the axle end in dotted ones.

FIG. 38.



In addition to these, two other diagrams were taken in the following manner:—Without removing the card from the barrel of the indicator the string was disconnected from the piston rod and attached to the slide rod. The indicator barrel now received its motion from the slide rod, and was quite disconnected from the piston. The indicator was then worked precisely as in the ordinary manner, connection being first made between it and one end of the cylinder and then the other. The result was two curious looking figures with a loop at the top of each. The slide card belonging to the cover end is shown in full black lines; that belonging to the axle end in dotted ones. These cards enable us to determine accurately the point of cut off on the ordinary card.

If we observe the point A on the slide diagram, a little consideration will show that it corresponds to the point A′ on the ordinary or piston-rod card. It is at this point that the slide opens and a sudden inrush of steam takes place, and the pencil of the indicator follows the line of the arrow. The slide opens wider and wider until it reaches its maximum opening, viz. D E, and then it begins to close, and when it reaches the point from which it started the steam will just be cut off, and the pencil of the indi-

cator will be at B, for to find the point of cut off on the slide diagram we have merely to draw a line at right angles to the atmospheric line through the point A, and where this line cuts the slide diagram again is the point of cut off on that diagram. If we measure from B down to the atmospheric line, we find the pressure in the cylinder above the atmospheric incline at the point of cut off, and if we rule B C parallel to the atmospheric line we know that the point C is the point on the card where the pressure is equal to the pressure at the instant of cut off. The point C is therefore the cut off. The point of cut off for the axle end may be found in the same manner. Inasmuch as the pulley on which the indicator string runs on the indicator is usually slightly larger than the barrel of the indicator, F G will not represent the whole travel of the slide perfectly, but this may be ascertained by measurement. Supposing it to be S, then $\frac{S}{FG} \times DE$ = actual steam opening at the cover end.

Again, the distance from A to A″ must be equal to twice the lap* if there is no inside lap on the slide, because the two steam openings (which are slightly different) + twice the lap, must be equal to F G, and deducting the two steam openings, viz. D E and the opening at the other end (not drawn in), we have A A″ left.

Divide this distance in two, rule a line perpendicular to the atmospheric line, and another through the point where this line cuts the slide card parallel to the atmospheric line, and this will give us the point in the piston or ordinary card where the exhaust took place, viz. H. If, however, there had been inside lap on the slide, it would only have been necessary to lay off from the point A towards A″, a distance equal to the inside + the outside lap, to the same scale as F G compared with the real stroke of the slide, viz. $\frac{FG}{S} \times \left(\text{inside + outside lap}\right)$.

Such slide cards are, however, of no good where there is an expansion slide on the back of the main slide.

* Supposing F G to represent the whole travel correctly.

CHAPTER II.

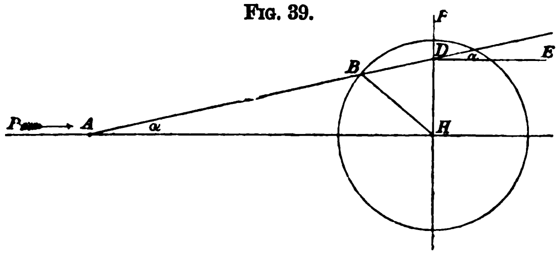
STRAINS ON THE AXLE.

The indicator card gives us the best guide to the strains upon the crank axle.

In considering this question, I shall in the first place neglect the weight of the moving parts, and take them into the calculation subsequently, when it will be found that they may very considerably alter the shape of the curve of strains.

And firstly, I will give a very simple method of making these calculations. Let H in Fig. 39 represent the centre of the axle, and H B the centre line of the crank at any portion of the

Fig. 39.



stroke, and A B the connecting rod in the corresponding position. Produce A B until it cuts the line H D, drawn at right angles to A H, at D. Then if P be the total pressure on the piston, as shown by the card, the torsional strain will be in all cases $P \times HD$. The proof is this. The force P acts in a direct line towards the centre of the axle, that is, along the line A H; therefore the thrust along the line A B is $P \times \frac{AD}{AH}$. We have therefore a force acting along the connecting rod $= P\frac{AD}{AH}$, and this force we may suppose to act at the point D. Now let us

resolve this force into one acting along the line D F, which will produce no strain tending to rotate the axle, and therefore we need say nothing more of it, and the other along the line D E; this latter is $\left(P\frac{A\,D}{A\,H}\right) \times \frac{A\,H}{A\,D} = P$, and $P \times D\,H$ is therefore the moment of the tangential force, or the force tending to rotate the shaft.

The friction has been neglected in the above, as well as the inertia of the moving parts and their weight.

In determining the rotary strain at different points of the stroke, all we have to do is to produce the centre line of the connecting rod until it cuts the line H F, and then multiply the number of units of distance in H D by the total number of units of pressure on the piston; taking pounds, inches, or any other quantities for the units, and remembering that the answer will be in these units. In the case of an oscillator the strain is slightly different. At each point it will be readily seen to be $P \times H\,D \times$ cosine of the angle of inclination of the centre line of the cylinder to the vertical centre line. In an oscillator the pressure acts, so to speak, along the connecting rod, that is, along the line A B and not along the line A H.

The first case which I shall take is that of an oscillating low-pressure engine, with cylinders 63 inches diameter and a stroke of 4 feet 6 inches. The distance from the centre of the trunnion to the centre of the axle was 9 feet.

The indicator card from the engine is drawn to such a scale as to be exactly the length of the stroke, and placed under it (see Fig. 40). The crank-pin circle may be divided into 20 equal parts, 1, 2, 3, &c., but more are better, and from them with centre O, which is the centre of the trunnion, are struck a series of circles. The distances of the points where these circles cut the line O C, from the point A, show what position the piston is in, at the points 1, 2, 3, 4, &c. If lines are ruled from them to the card below we see the pressures corresponding to these points. And then to find the moments of the tangential force we multiply $P_1 \times$ area of the cylinder $\times$ C D $\times$ cos. C O D, and $P_2 \times$ area of cylinder $\times$ C E $\times$ cos. C O E. We may, however, in practice omit to multiply each time by the cos. of the angle of inclination, because the greatest

error introduced is only 3 per cent. in this case, which is a fair sample of any ordinary oscillator.

Fig. 40.

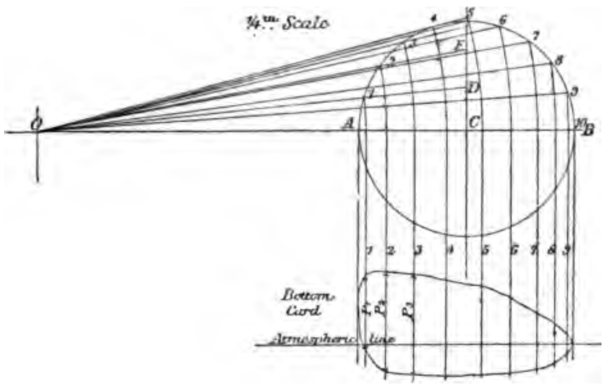
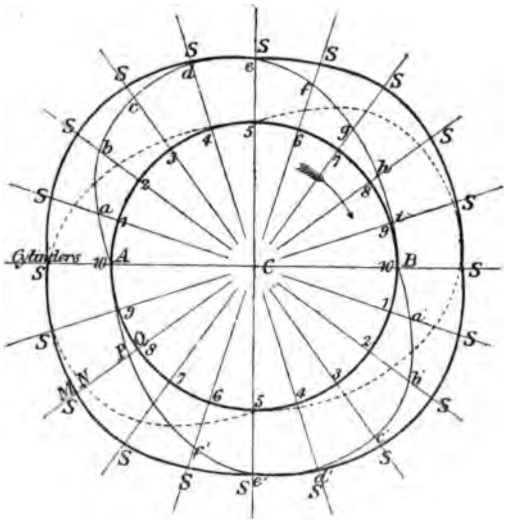


Fig. 41.



Having calculated the moments of the tangential forces at all the different points, 1, 2, 3, &c., the next thing (see Fig. 41, where the same points 1, 2, 3, &c., are shown) is to draw radial lines through these points, and then starting from the crank-pin circle, lay off distances outside this circle along these radial lines proportional to the moments. Joining all the terminal points, we get a

curve of strains, a, b, c, d, e, &c., for one cylinder on the up stroke. The card for the down stroke must now be worked out in the same way, and we get the curve a', b', c', d', &c.

In a screw engine the shaft has to transmit the torsional strain of both cylinders continually from the engine-room to the screw; but in the case we are considering, a paddle engine, there is a wheel to each cylinder, although when the ship rolls one wheel may be doing a very little work for a short time. If the one wheel were to come out of the water entirely both cylinders would strain the same piece of shafting, and as they take on to cranks at right angles we should proceed in the following manner to find the curve of strains on that piece of shafting.

On a piece of tracing paper make a tracing (see Fig. 41), showing the two curves a, b, c, d, e, &c., and a', b', c', d'. Then keeping the centre C on the tracing paper, over the centre C, turn the tracing paper through 90°, or a quarter of a circle,* and then prick off the curves through the tracing paper on to the figure below. This gives us the dotted curves, and they represent the strain of the second cylinder on the shaft, while the curves a, b, c, d, &c., represent the strain of the first. In order to ascertain the total strain we must now add together the strains of each, and we get the curve S, S, S, S, &c. Thus $MQ = NQ + PQ$. The curve S, S, S, represents the total moments in whatever units we may have adopted of tangential force of both cylinders at all portions of the revolution, supposing the whole power to be transmitted to one wheel.

From this curve we see what the maximum strain is which the shaft will ever have to bear so far as steam pressure is concerned, and if the ideal card for the engine had been used we might make use of this knowledge in determining the proper diameter of the shaft.

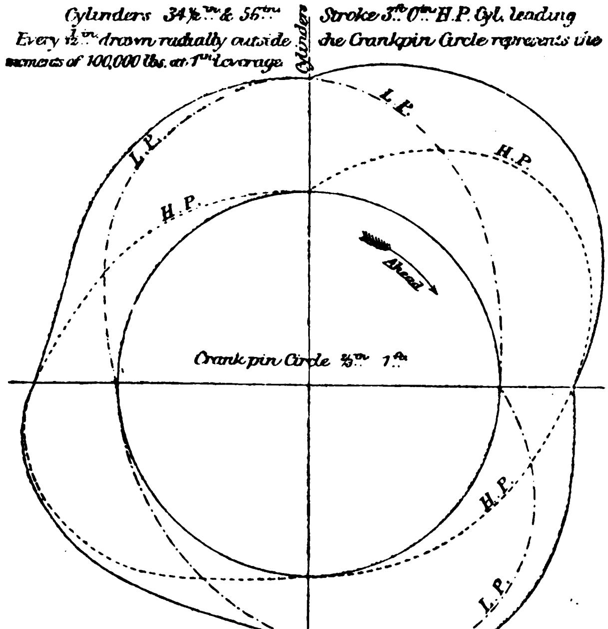
We see on the curve S, S, S, that the distances measured radially from the points S, S, S, to the crank-pin circle do not vary very much; or, in other words, the force tending to rotate the axle is moderately uniform; the smallest distance being to the largest in the proportion of 80 to 100.

In Fig. 42 is given the curve of strains on a compound-engine cylinders $34\frac{1}{2}$ and 56 inches, 3 feet stroke, cutting off at close

* Or whatever number of degrees one crank is in front of the other.

upon $\frac{3}{10}$ths in the high-pressure cylinder. The greatest twisting strain on the crank in this case is to the smallest as 100 to 37. We see that in this case the variations of strain on an axle of a compound are considerably greater than those on the axle of a low-pressure oscillating engine. On Figs. 43 and 44 the cards are shown in full black lines from which the strains were calculated.

FIG. 42.



On Fig. 45 is the ideal card $p \times v = c$ of a two-cylinder engine, 39 inches diameter, 3 feet stroke, running 67 revolutions. The piston speed of this engine is the same as that of the compound just considered, the indicated horse-power is also the same, and the same weight of steam at the same boiler pressure is used per indicated horse, that is, supposing there to be the same amount of condensation in the cylinders. The tangential strains produced by such a pair of cylinders are shown on Fig. 46, and comparing that figure with the one for the compound, it will be seen that the

maximum strains are nearly the same, and the minimum strains not very different.

FIG. 43.

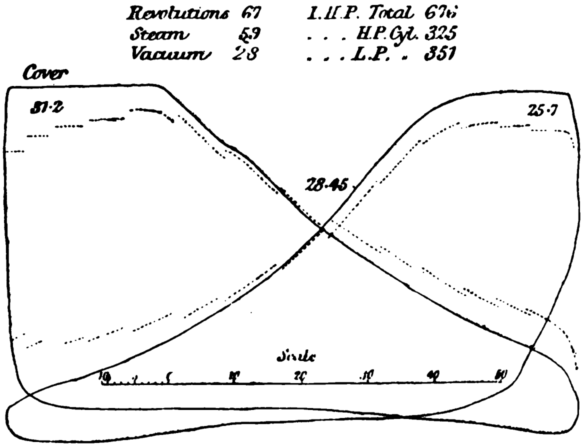
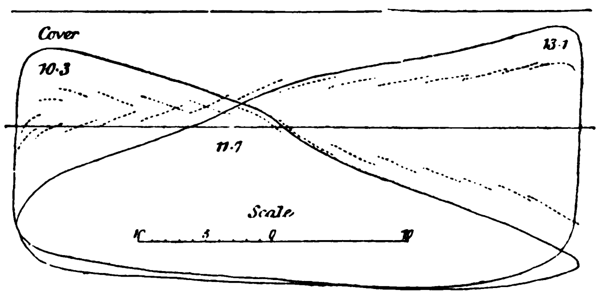


FIG. 44.



The question which now remains is to observe how the strains on the shaft are modified by the inertia and by the weight of the moving parts.

For this purpose we shall consider the curve in Fig. 42 produced by the compound engine before referred to on page 37.

The piston rods with brasses weighed each about 9·4 cwt., or 1050 lbs., being the same for the high and low pressure cylinders. The connecting rods weighed each about 10·5 cwt., or 1180 lbs. The high-pressure piston weighed 7 cwt., or 784 lbs., and the low-pressure piston weighed 15·5 cwt., or 1740 lbs. The total weight of the high-pressure piston, piston rod, and connecting rod

is therefore 3014 lbs., and the total weight of the low-pressure piston, piston rod, and connecting rod is 3970 lbs.

FIG. 45.

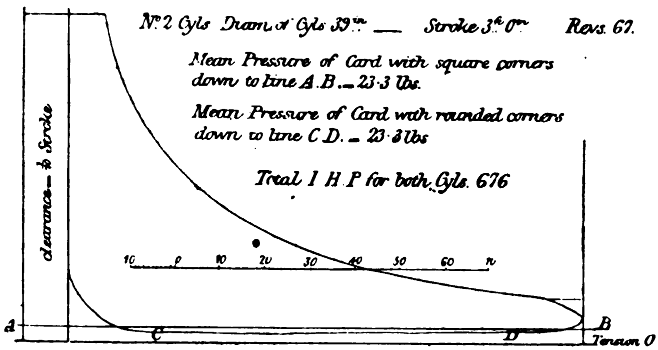
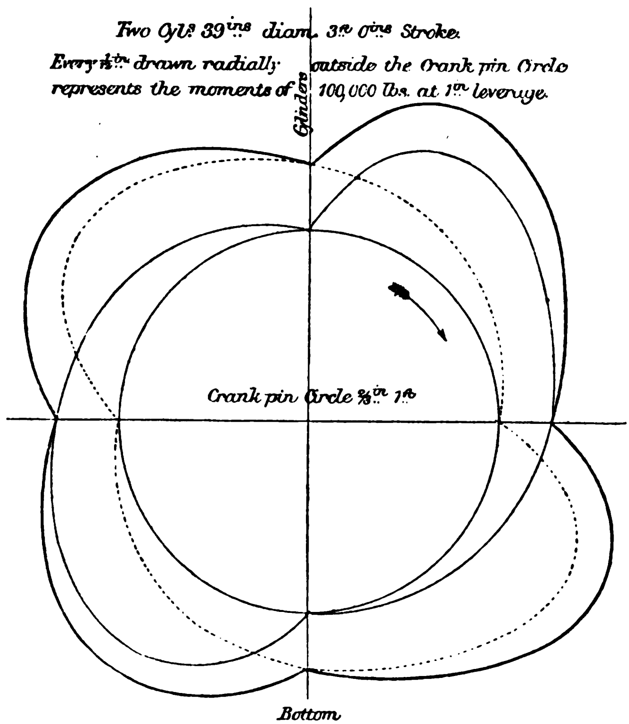


FIG. 46.



The engine in question was of the inverted direct-acting type, and in considering the curve of strains it is clear that we have to

deal with three separate quantities: first, the strains produced by the steam, the curves of which we have already laid out; secondly, the inertia of the moving parts which has to be overcome four times each revolution for each cylinder: that is to say, they are put in motion during the first half of the down stroke; then that motion must be destroyed; then they are put in motion on the up stroke, and again the motion has to be destroyed; and thirdly, we have in the case of a vertical engine to take into account the dead weight of the moving parts if there are no counter-balance weights on the shaft, this dead weight acting against the steam on the up stroke, and with it on the down stroke, whatever the piston speed of the engine may be. The counter-balance weights destroy the torsional strain produced by the weights of the moving parts, but double the transverse strain, and introduce other objectionable forces. Let us firstly consider the weights.

In this engine the high-pressure piston, piston rod, and connecting rod weighed 3014 lbs., and the area of the cylinder was 935 inches. On the up stroke we have a weight of about $3\frac{1}{4}$ lbs. on each square inch of the high-pressure piston acting against the steam, and on the down stroke acting with it. In order, therefore, to get at the actual torsion curve, the steam line of the cover card must be raised $3\frac{1}{4}$ lbs. all along it, and the steam line of the axle end card must be lowered that amount. And the same process should be gone through with the low-pressure cards, and the cards so altered are corrected for the weight of the moving parts. This is so simple that I have not given the curve produced in this way. The next question is the effect produced by the momentum or inertia of the moving parts.

The first thing is to see how the velocity at which a body is travelling and its momentum (used here in its ordinarily understood sense of vis viva) are connected. The momentum of a body depends solely upon its weight and upon the velocity at which it is travelling.

If a weight of 10 lbs. is at any moment travelling at a velocity of 100 feet a second, whatever direction it is moving in its momentum at that moment is precisely the same as it would have been after falling through the height necessary to give it this velocity. The two weights are the same, and so are the two velocities, and therefore the momentum is the same in both.

So that whenever we have a weight moving at any velocity we can say that its momentum is that of a certain weight falling through a certain distance, and the momentum of all moving bodies can be referred to this standard. But a body falling through a certain distance represents, or is equivalent to, a certain number of foot-pounds, as referred to before when speaking of indicated horse-power on pp. 3 and 4. So that the momentum of every moving body can be represented by speaking of it as so many foot-pounds: by which we understand that the energy expended in imparting a velocity to a weight, or in destroying that velocity, is exactly equal to the energy required to raise a certain number of pounds one foot high. And we see at once the error occasionally entertained that the momentum of a body has an equivalent in dead weight. The blow of a steam hammer has an equivalent in foot-pounds, but none in dead weight.

Now elementary dynamics show us that if V is the velocity at which a body travels, and h the height through which it must fall to gain that velocity, and t the time during which it is falling, then

$$h = \tfrac{1}{2} g t^2, \qquad \text{and } V = \sqrt{2 . g . h}, \qquad \text{and } V = g . t,$$

Where g represents the force of gravity, which is a uniformly accelerating force giving a velocity of 32·2 feet a second to falling bodies: for g we may write 32·2 in the above equations.

Now applying these formulæ to the case of a weight travelling with a velocity V, W × h is the measure in foot-pounds of the body's momentum (W being the weight and h the height through which it has fallen); but since

$$V = \sqrt{2 . g . h}, \quad \therefore V^2 = 2 . g . h \quad \text{and} \quad h = \frac{V^2}{2 . g},$$

therefore by substituting we get

$$W \times h = W \frac{V^2}{2 . g}.$$

This shows us that the momentum of a body varies as the square of the velocity.

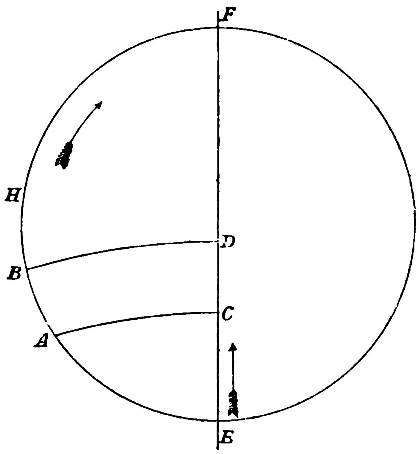
Suppose, for example, a weight of 60 lbs. travelled with a velocity of 30 feet a second, the momentum is $\frac{60 \times 30^2}{64 \cdot 4} = 835$ foot-pounds.

Having found a simple formula which will give us the momentum of a moving weight in foot-pounds, we will return to the engine.

In making the necessary calculations I shall assume what cannot be proved to be correct, but which is so very nearly, viz. that the rotation of the crank pin is uniform, and, further, I have neglected the movement from side to side of the correcting rod.

Consider two points, A and B, on the crank-pin circle near together, Fig. 47. While the crank pin travelled from A to B the piston, &c., moved through a certain distance from C to D. At the

Fig. 47.



instant when the crank pin passed A, the piston, &c., were moving at a certain velocity along the line E F, and if we call this velocity V_1, then the momentum of the piston at the point A is $W \frac{V_1^2}{2 . g}$ where W represents the weight of the piston, piston and connecting rods. When the crank pin is at B the velocity is now V_2 along the line E F, and the momentum of the moving part is now $W . \frac{V_2^2}{2 . g}$.

Since $\frac{W}{2g} V_1^2$ was the momentum of A, and $\frac{W}{2g} V_2^2$ was the momentum at B, the momentum added to or taken from the piston in passing from C to D is necessarily $\frac{W}{2g} (V_2^2 - V_1^2)$. If, then, we

can calculate the velocities at these two points, we can then calculate the difference in the momentum.

Momentum is, as we have seen, measured in foot-pounds; therefore $\frac{W}{2g}(V_2^2 - V_1^2)$ is an expression for a certain number of foot-pounds which have been expended between C D, C D being expressed in feet. Let P be a pressure or weight raised through a distance C D, then P × C D represents the foot-pounds, and if the amounts are made equal

$$P \times CD = \frac{W}{2g}(V_2^2 - V_1^2),$$

and therefore

$$P = \frac{W(V_2^2 - V_1^2)}{2 \,.\, g \,.\, CD}.$$

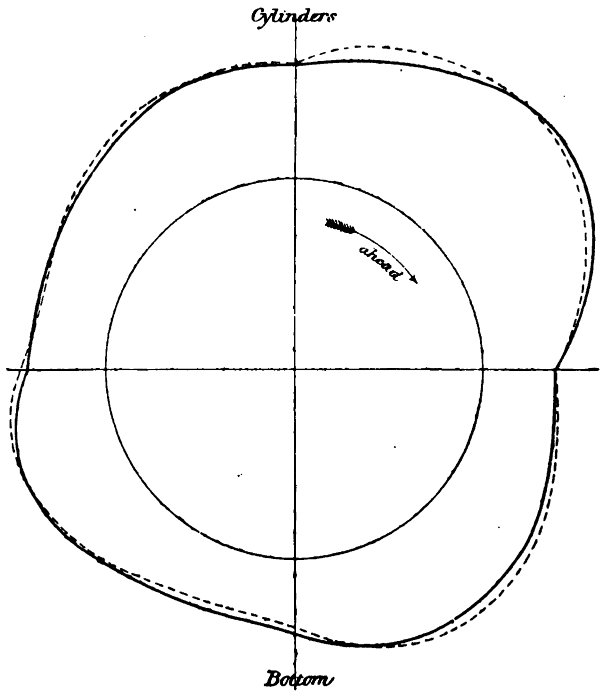
That is to say that a pressure, P, moved through a distance C D represents an amount of energy equal to that which the piston, &c., have gained in passing from C to D; and from the work done in turning the shaft between these two points that amount of energy must be deducted. If we divide P by the number of square inches in the piston we get at a pressure per square inch which carried through the distance C D is equal to the momentum acquired by the piston. From the pressure as shown by the indicator card we must deduct this new pressure. After this has been done a new point, H, on the crank-pin circle is taken, and the pressure to be deducted between B and H found, and so on round the half circle. The dotted lines in Figs. 43 and 44 show the pressure to be deducted in this particular instance.

We have deducted a series of uniform pressures between C and D, &c., but, in fact, the pressures are not uniform; we have deducted or added, as the case may be, the proper amounts, but not precisely in the proper way.

We know that if the pressures to be deducted are not uniform, being less than our main pressure, as shown by the dotted lines at one point, and greater at another, the lines marking such pressures will cut each of the short dotted lines somewhere, and we may by joining the centres of these dotted lines get a card which will very nearly represent the pressure straining the axle torsionally. This has been done in this case, and Fig. 48 shows in full black lines

the strain produced by such a corrected card, and the dotted black lines show the strain of the original cards, as already shown in Fig. 39.

Fig. 48.



Although the strains, which are not shown, for each cylinder considered individually are much altered, those for the two cylinders together are not so. It may be noticed that the mean pressure of the corrected card must remain the same as that of the original one. Any momentum acquired by the pistons at the beginning of their strokes, and represented in the indicator card by a deduction of pressure, is returned to the shaft towards the end of the stroke when this momentum is destroyed or transmitted through the connecting rod, where it is represented by an increase of pressure. In fact, the momentum of the parts does not in any way (excepting friction) affect the whole horse-power, merely altering the manner in which it is exerted. It will be remembered that no correction was

made for the dead weight of the moving parts; see p. 39. The engine has in fact been treated as if it had been a horizontal one.

We now must see how the velocities of the piston, &c., at the different points may be found. This may perhaps be done as quickly and more safely by drawing out the connecting rod and crank-pin circle to a large scale, but it may easily be calculated as follows:

In Fig. 49 let A B, or C, be the connecting rod, let the throw be represented by R, and let the distance A O be represented by x. Plane trigonometry tells us that

$$C = \sqrt{x^2 + R^2 - 2\,R\,x\cos. a},$$

$$\therefore\ C^2 = x^2 + R^2 - 2\,R\,x\cos. a,$$

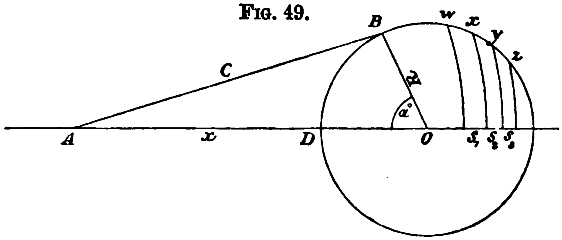
$$\therefore\ x^2 - 2\,R\,x\cos. a + R^2 - C^2 = 0.$$

This is a quadratic equation, the solution to which is

$$x = \frac{2\,R\cos. a + \sqrt{\{(2\,R\cos. a)^2 + 4\,(C^2 - R^2)\}}}{2}.$$

And it must be remembered that when a becomes greater than 90° its cosine becomes negative, making 2 R cos. a negative, but leaving its square positive as before.

Fig. 49.



This is an equation from which x may easily be found corresponding to different values of a. At the very beginning of the stroke x was equal to C + R, and if we deduct the new values of x corresponding to the various values assigned to a, from this constant C + R, we get the distances which the piston has travelled from the beginning of the stroke, when the crank is at the various angles a, which may be 10°, 20°, 30°, and so on to 180°.

Now looking at the four points $w\,x\,y\,z$ at equal distances on the circumference of the crank-pin circle: if we deduct the distance

the piston has travelled from the beginning of the stroke at W from the distance it has travelled over at x, we get the space marked S_1, similarly we find spaces S_2, S_3, &c. The first thing then is to calculate all the spaces S_1, S_2, S_3, &c. Now if t be the time in fractions of a second occupied by the crank in revolving from w to x, x to y, &c., the velocity at the point x of the piston (not the crank pin) may be considered as being $\frac{S_1 + S_2}{2 \times t}$. This is only an approximation, but approaches nearer and nearer to the truth as the distances $w\,x$, $x\,y$ are made smaller. If S, &c., had been calculated in inches, the velocity in feet at the point x would be nearly $\frac{S_1 + S_2}{12 \times 2 \times t}$, and similarly the velocity in feet at y may be considered as $\frac{S_2 + S_3}{12 \times 2 \times t}$, and therefore the momentum gained by the piston in passing from x to y, that is, through the space S_2, or lost, according as there is an increase or a diminution of velocity, is according to our formula $\frac{W}{2g} \times \left(V_2^2 - V_1^2\right)$

$$\frac{W}{2g}\left\{\left(\frac{S_1 + S_2}{12 \times 2 \,.\, t}\right)^2 - \left(\frac{S_2 + S_3}{12 \times 2 \times t}\right)^2\right\};$$

and if P were a uniform pressure moved through a distance in feet of $\frac{S_2}{12}$, then

$$P\frac{S_2}{12} = \frac{W}{2g}\left\{\frac{(S_1 + S_2)^2 - (S_2 + S_3)^2}{(12 \times 2 \times t)^2}\right\},$$

and

$$\therefore P = \frac{W}{2 \times 32{\cdot}2 \times 12 \times 4 \times t^2} \times \frac{(S_1 + S_2)^2 - (S_2 + S_3)^2}{S_2}.$$

If, as in the case considered, the distances $w\,x\,.\,x\,y\,.\,y\,z$ had been taken corresponding to 10°, the spaces S_1, S_2, S_3, &c., are easily calculated from the formula given, by logarithms, and the subsequent calculations are mere numerical ones.

If $\frac{W}{2 \times 32{\cdot}2 \times 12 \times 4 \times t^2}$ be calculated and represented by C, then

$$P = C\,\frac{(S_1 + S_2)^2 - (S_2 + S_3)^2}{S_2}.$$

P is thus calculated for each of the spaces S_1, S_2, S_3, S_4, &c., be-

ginning at the beginning of the stroke. It represents the uniform pressure which when moved through spaces S_1, S_2, S_3, &c., equals the momentum gained or lost, as the case may be, by the piston, &c., while moving through the same spaces. If these uniform pressures be divided by the number of square inches in the piston we get a uniform pressure per square inch to be added or deducted from the steam pressure as shown on the card through the spaces S_1, S_2, S_3, &c. See Figs. 43 and 44.

CHAPTER III.

FUEL.

Whenever fuel of any description, either wood or coal or lignite, or in fact when any body, as tallow, grease, oil, or coal gas burns in the open air either quickly or slowly, that body combines with a certain portion of the air, and forms a chemical compound. This compound is generally a gaseous one at first, but does not always remain so. When charcoal burns, the resultant product is entirely gaseous; when coal burns, the large proportion of the resultant product is gaseous, a certain portion is not so. When a piece of iron is burnt, the scales which are the resultant product are altogether solid, there is no gaseous matter formed.

A large amount of heat is generated whenever a body burns, and a constant amount of heat is always generated when the same substance is burnt under the same conditions, and the same chemical products are the result. Thus, you cannot burn a ton of coals one day and generate a certain amount of heat, and burn another ton of the same coal on another but similar day, and get a different result. It must be remembered that each particular sort of fuel, when completely burnt, will generate a certain constant quantity of heat, no more and no less. This must not be understood to mean that in all cases a ton of any fuel, as coal, will invariably evaporate the same quantity of water in a boiler, but merely that in burning it invariably gives out the same amount of heat. Whether the same proportion of the whole heat given out is in all cases absorbed by the water in the boiler, is another matter, depending very much upon the rate at which the coal is burnt, and the order in which the fires are kept. If the fires are pushed, the gases escaping up the chimney become hotter, and there is a consequent waste of fuel; or again, if the fires are allowed to burn into holes, a large quantity of cold air rushes through the bars, without being burnt, and cools the boiler.

I will now endeavour to show, firstly, what the chemical compounds are which are formed when coal burns, and secondly, what the constant amount of heat given out by burning carbon is, what relation this bears to the work done by an engine, and what the relation should be if the engine and boiler were perfect machines.

The atmosphere has been ascertained to be composed of two elementary colourless gases, nitrogen and oxygen, in the proportion by weight of 76·9 of nitrogen to 23·1 of oxygen, and by volume in the proportion 79·1 of nitrogen to 20·9 of oxygen.

These gases are called elementary because they cannot be decomposed, or resolved into any simpler gases. They are not the result, so far as is at present known, of the combination of any other gases. They are ready to form compounds with other elements under favourable circumstances, and these compounds may be again decomposed into the same elements, but there the decomposition of the compound, or more properly, the disuniting of the elements, stops.

Of these two elementary gases which together form the atmosphere, only one, viz. the oxygen, combines with the coal when it burns. The other, the nitrogen, is merely heated by coming into contact with the burning coal and the products of the combustion, but enters into no combination.

The amount of either oxygen or air required to effect the complete combustion of a pound of coal is known. It has been determined by chemists by experiment, that when 12 lbs. of pure dry carbon or charcoal (which is by far the largest constituent part of coal which burns), burn completely in the air, they unite with 32 lbs. of oxygen, and therefore when 1 lb. of such substance burns in the air, it unites or combines with $\frac{32}{12} = 2 \cdot 667$ lbs. of oxygen, and since oxygen is only the $\frac{23 \cdot 1}{100}$ part of the atmosphere by weight, therefore 1 lb. of carbon requires $2 \cdot 667 \times \frac{100}{23 \cdot 1}$ lbs. $= 11 \cdot 545$ lbs. of air to effect its complete combustion.

The product of this combination of a solid and a gas is a gas called carbonic acid, and whenever carbon is completely burnt in the air, this gas is formed.

We see that whenever carbon is completely burnt, we have two hot gases going up the chimney, viz. the carbonic acid, and the nitrogen of the air.

The ordinary meaning of the term gas as applied to coal gas for illuminating purposes is a very restricted, narrow one; chemists apply the term to all substances which are neither liquid nor solid, with a qualification to be noticed later on, and it is in this wider sense that the word has been used above. Thus the suffocating fumes from burning sulphur, ammonia, carbonic acid, sulphuretted hydrogen are all gases.

If, while the carbon was burning, an insufficient supply of air had been allowed to it, an incomplete combustion would have taken place, and some of the carbon, instead of burning so as to form carbonic acid, would unite with only one-half of its proper allowance of oxygen, and would form a gas called carbonic oxide. A combustion is said to be incomplete when the burning body, which is coal in this case, has not been able to unite with as much oxygen as it is capable of uniting with. The reason why a certain portion of the carbon would unite with only $\frac{1}{2}$ and not $\frac{5}{8}$ or $\frac{3}{4}$ of the whole amount of oxygen it could unite with, will be found in all chemical works, under the Atomic theory. This carbonic oxide will, however, burn when it comes in contact with fresh air, provided it has not been too much cooled, and will then form carbonic acid.

When flames are seen coming out of the top of a chimney of a marine boiler, it is chiefly this carbonic oxide which is burning, not the flame of the fire below which reaches up that height; this flame is composed partly of other gases, as will be seen later on. We may be sure that whenever carbonic oxide is so formed that the coal is being burnt at a great loss, a great quantity of the heating power is being sent up the chimney. This loss is far greater than one might at first be inclined to imagine, for the amount of heat given off by carbon in burning to its first stage, viz. to carbonic oxide, is not nearly so great as that given off in burning to its second stage, viz. to carbonic acid. I shall refer again to the subject of carbonic oxide when I come to speak of the amount of heat developed by burning fuel.

It is important to remember that when coal or carbon burns, carbonic oxide is first formed, and if a sufficient supply of air is

admitted, the carbonic oxide unites with a second portion of oxygen equal in weight to the first, and instantly forms carbonic acid.

We have up to the present moment considered what the products are when the pure element, carbon, burns in the air, but as a matter of fact we seldom or never have to deal with anything but mixtures of pure carbon with other bodies. Coals, for instance, are never pure carbon, and more than this, no two coals ever have precisely the same composition. Coke is not pure carbon nor is charcoal, although much more nearly so than either coal or coke. Wood again contains all sorts of things besides pure carbon.

The following is a list of some common coals and their compositions, from which it will be seen that they all contain mineral matter, and more or less gaseous matter also; by gaseous is meant that the matter is gaseous in its elementary form.

The figures have been averaged from those given in Dr. Percy's 'Metallurgy.'

TABLE SHOWING THE APPROXIMATE COMPOSITIONS OF VARIOUS BRITISH COALS (the principal constituents only being mentioned).

		Carbon.	Hydrogen.	Oxygen.	Ash.
CAKING COAL ..	Northumberland ..	80·3	5·3	10·7	1·7
	Nottingham	77·4	5·	7·8	4·
	Blaina, S. Wales ..	83·	5·7	6·2	2·6
NON-CAKING COAL.	South Staffordshire ..	74·	4·8	..	2·6
	Ayrshire and Fifeshire	79·3	5·	..	1·4
	Dowlais	86·5	4·4	..	3·4
CANNEL COALS ..	Wigan	82·	5·6	..	2·6
COKE	.. ..	89·6	..	..	8·
ANTHRACITE ..	South Wales	91·5	3·3	2·7	1·6

NOTE.—"In calculating the heat which a coal will generate, that hydrogen only which is in excess of the amount required to combine with the oxygen in the coal is to be considered as available."—Authority: PERCY.

All those constituents of coal which will combine with oxygen are valuable as fuel and will generate heat, and all those which have already combined with as much oxygen as they will unite with are useless for that purpose. The products formed by the combustion of all these different elements are very various, but for practical purposes, so far as the heat generated is concerned, the great fact is

that the carbon forms either carbonic acid, or oxide, and that the hydrogen unites with oxygen to form the vapour of water.

To determine the theoretical evaporation of coal, we must ascertain how much of its constituent parts are capable of uniting with oxygen, and then see how much heat is generated when a pound of each of these bodies burns. This theoretical evaporative power of coal is not, however, a good criterion of its steaming qualities, for these do not depend upon the chemical composition of the coal only. Thus one coal may cake too much, another may require too strong a draught, a third may crumble very much and go to waste with the ashes. Some coals again, and wood especially, contain a large quantity of water, and part of the heat generated by them in burning, goes to the evaporation of it. Other coals contain a large quantity of ash, which prevent their being very good for steaming purposes.

It has been determined experimentally that carbon in burning from the solid condition to carbonic acid generates 14,544 units of heat, and carbon in burning to form carbonic oxide generates 4452 units. That is to say, one pound of carbon in burning to form carbonic acid generates heat sufficient to raise 14,544 lbs. of water 1° Fahr. If then it generates 14,544 units of heat in burning to carbonic acid, and only 4452 units of heat in burning to carbonic oxide, it follows that carbonic oxide, containing 1 lb. of carbon, would generate in burning to carbonic acid 10,092 units: so that carbon in burning through its first stage to carbonic oxide gives off only 30·6 per cent. of that which it gives off in completely burning through both the first and second stages to carbonic acid.*

We see then the necessity of admitting a sufficient supply of air to the coal and burning, not allowing any of it to escape in the form of carbonic oxide, and the prodigious unnecessary waste which takes place when carbonic oxide is allowed to burn in or at the top of the chimney instead of in the furnaces.

Nearly all coals, more especially the caking ones, contain considerable quantities of gaseous matter, or more properly, a considerable quantity of matter which is in its elementary or uncombined condition in the form of a gas. When such coal is placed in a

* Hydrogen in burning to form water generates 62,032 units F.

retort and heated from the outside, as is the case when coal gas is made, the water and a large quantity of gaseous matter is distilled off, and this same distilling action takes place in the furnace of a boiler. The gases so formed escape unburnt if an insufficient amount of air is admitted, and as in the case of the carbonic oxide, if they are not too much cooled in passing through the tubes of the boiler, they take fire when they get through them and come in contact with air, or when they come to the top of the chimney. Here again we see the importance of a sufficient supply of air. And we may observe in passing that one very bad effect of small boiler tubes, where the draught is not strong, and where the gases consequently pass through them slowly, is to cool them and put out the flame before the combustion is complete.

Smoke is carbon chiefly, in an extremely finely divided condition, and is of course, weight for weight, a dead loss of fuel to the stoker, who does not get even a fraction of the total heat, as in the case of carbonic oxide.

We have now seen what are the chief products of combustion of coal, and what the amounts of heat are which carbon gives out in burning.

I shall now proceed to the consideration of what the relationship existing between the heat developed by the coal and the work done by the engine is, and what the relationship should be with perfect apparatus.

Two very distinguished men, both living, one an Englishman, Joule, the other a German, Mayer, established by two very different methods the law that the same amount of heat is always equivalent to the same amount of mechanical work; and conversely, that the same amount of mechanical energy will, when turned into heat, always produce exactly the same amount of it.

They ascertained what the corresponding amounts of heat and work are, and by so doing opened the eyes of engineers to the prodigious waste which is going on in every steam engine, and in fact almost everywhere where fuel is used. I shall return to both Joule and Mayer's methods later on in the chapter upon the expansion of gases; but at present I prefer to lay stress on Joule's method of approaching the subject.

Dr. Joule, of Manchester, had a vessel containing water, in

which he could cause to rotate round a vertical axis a series of paddles. At the top of this vertical spindle he fixed a horizontal pulley, and round it wound a long string, fastening one end to the pulley, and leading the other end over a second pulley placed vertically to a weight, to which the string was attached. So long as the weight was stationary the paddles were perfectly still; but the moment the weight began to fall the paddles were constrained to revolve, and continued to do so until the weight was stopped. During the movement of the paddles the water was violently agitated, and the effect of this churning was to raise the temperature of the water slightly. Joule found that this rise of temperature depended upon the height through which the weight had fallen, and upon the amount of that weight. He found, after making a very large number of experiments, conducted in a great variety of ways, that a weight of 1 lb. falling through 772 feet would raise 1 lb. of water through 1° Fahr.; or, what is just the same thing, a weight of 772 lbs. falling through 1 foot. The number 772 is the one now generally accepted, and this number is what is called the mechanical equivalent of heat; but in stating the mechanical equivalent it is important to remember what units it is expressed in. In this case the units are foot-pounds and degrees Fahr.

The knowledge which this law of Joule's gives us is exceedingly valuable. From it we learn that in the very best engines that can be made, we are only getting a most insignificant part of the whole power out of the coal which there is in it. And in the author's opinion it shows the futility of attempting to effect any very great saving in coal by means of new arrangements of slide gear, and the whole host of other small matters which do no doubt in a measure contribute to the efficiency of an engine, but never can become radical improvements.

Supposing a vessel to have performed a voyage during which the amount of coal burnt has been carefully noted, and that indicator cards have been taken at regular intervals from the engines, so as to enable us to say what the average indicated horse-power has been. We will take an engine averaging 2500 horse-power, and burning on a ten days' voyage 600 tons. This would be equivalent to a consumption per hour of 5600 lbs. And the con-

sumption per indicated horse-power per hour would be $\frac{5600}{2500}$ $= 2\frac{1}{4}$ lbs. very nearly.

As we have seen before, some of the coal is not of any value, a portion being incombustible. If we turn to page 48, we shall see that the anthracite contains in round numbers only 90 per cent. of carbon. If we neglect the other constituents, we may consider that, instead of burning $2\frac{1}{4}$ lbs. of coal per indicated horse-power per hour, we are burning 90 per cent. of $2\frac{1}{4}$ lbs. of pure carbon, or just about 2 lbs.

We know that a pound of carbon generates, while burning to carbonic acid, 14,500 units of heat, that is, gives off as much heat as will raise 14,500 lbs. weight of water 1° Fahr., and therefore 2 lbs. will generate 29,000 units.

We are therefore generating in round numbers 29,000 units of heat, and getting in exchange 1· indicated horse-power.

We have just seen that one unit of heat is equivalent to 772 lbs. raised 1 foot high, and therefore 29,000 units of heat are equivalent to 22,400,000 foot-pounds. But an indicated horse means 33,000 lbs. raised 1 foot per minute, which is equivalent to $33{,}000 \times 60$ lbs. raised 1 foot high per hour $= 1{,}980{,}000$ foot-pounds.

We see then that we are burning coal sufficient to raise 22,400,000 foot-pounds, and are actually raising 1,980,000. We are, in fact, out of a very good engine getting about one-eleventh of the power we should do.

No improvement which has been made to the steam engine since the time of Newcomen, in the first quarter of last century, would be in point of importance at all comparable to that which we can only hope for, which shall enable us to get even one-third of the theoretical mechanical equivalent of the heat of the coal.

When we see this tremendous difference between the work we should have got out of the coal, and the amount we really did get, we are naturally lead to reflect upon the causes, and to endeavour to discover where the greater part of the loss takes place. Such consideration must precede any rational attempt to effect radical improvements in the steam engine.

We will begin at the beginning, and follow the water in

the boiler through its different stages out of the boiler into the engine, through the engine, and out of it into the condenser.

When cold water is first heated the temperature goes on rising continually until it begins to boil, and when that point is reached there is no further rise in the temperature until the whole of the water has been evaporated. During the whole of this time vapour keeps rising from the water, and when once it boils the vapour comes off very rapidly. Vapour comes off water at all temperatures, and even rises off ice.

Boiling begins when the bubbles formed at the bottom of a vessel rise without condensation to the top; but as water usually contains a large quantity of air dissolved in, and is able to hold less in solution as it gets warmer, so, as the temperature rises a great number of air bubbles are formed, and rise to the top without condensation or being redissolved, but this is not boiling. When the steam is formed at the sides and bottom of the vessel, and in the body of the water as well as on the surface, then boiling has begun. When bubbles of steam begin to form at the bottom, and rise without condensation to the top, we may be sure that the tension or, as engineers would call it, the pressure of the steam inside the bubble must be exactly equal to the pressure outside it. If this were not the case, if it were either greater or less, the bubble would begin to expand or be compressed. The pressure on the outside of the bubble is the pressure of the air plus the pressure due to the head of water over the bubble. This latter is however small, as compared to the pressure of the air.

If, then, the pressure of the air over the water were diminished, by carrying the vessel up a mountain, the pressure inside the bubbles becomes correspondingly less, and boiling goes on at a lower temperature; and if the pressure had been increased, the temperature of boiling would have risen, for the pressure of the steam and the temperature are inseparably connected, that is to say, whenever the water boils at any particular pressure it will always be at the temperature due to this pressure: the higher the temperature the higher the pressure.

Tables III. and IV. show the pressures and the corresponding temperatures of steam.

If instead of the water having been boiled in a vessel open to

the air we had boiled it in a close boiler, the steam would not have been able to escape, and the pressure would have risen continually until the safety valves had lifted. The steam should then escape as rapidly as it is formed, and boiling would go on regularly at the temperature corresponding to the tension at which the valves blow.

And I may take the present opportunity to speak of the difference between the tension of steam and the pressure, a matter which is not always very clearly understood, although the two things are perfectly different. If you had a valve of exactly 1 square inch area placed on the top of a boiler, as a safety valve is, and loaded with a weight of 60 lbs., this valve would begin to blow when there was a pressure of 60 lbs. in the boiler; but the tension of the steam in the boiler would be 60 lbs. plus whatever weight the atmosphere pressed on per square inch. If this were 14½ lbs., then the tension of the steam would be 74½ lbs. If the whole boiler were surrounded by a vessel from which all the air had been pumped, it is clear that we should then require to load the valve with an extra 14½ lbs., instead of the atmosphere, bringing the total weight required to balance the steam up to 74½ lbs. The tension of the steam is the thing to look to chiefly; it always corresponds to a certain temperature. The pressure does not do so exactly, owing to changes in the weight of the atmosphere; for instance, 60 lbs. steam may sometimes be of a tension of 75 lbs., at other times of 74 lbs.

Returning now to the boiler, which we shall suppose to be worked at 60 lbs. pressure, corresponding approximately to a temperature of 307° Fahr., and that the feed water is pumped into the boiler at 132° Fahr., if we examine Table V. (which will be explained in the chapter on gases), we shall see that steam at the temperature 307° Fahr. contains 1176 units of heat. By this is meant that if you pass 1 lb. weight of steam at 60 lbs. pressure into water at 32° Fahr., the steam in condensing will give out heat sufficient to raise the temperature of 1175 lbs. of the cold water through 1° Fahr. To turn 1 lb. weight of feed water at 132° Fahr. into a pound weight of steam at 60 lbs. pressure, we must put in 1176 − 100 * = 1076 units of heat.

* 132° Fahr. are 100° above the freezing point.

Anthracite contains approximately 90 per cent. of carbon. One pound of carbon generates in burning 14,500 units of heat, and therefore a pound of anthracite generates 90 per cent. of 14,500 units, or 13,000. But if it only evaporates 8 lbs. of feed water, and each of these pounds when so evaporated contains 1076, then we are getting only $8 \times 1076 = 8600$ units, instead of 13,000. And therefore we see that for each pound of anthracite so burnt, 4400 units of heat go straight up the chimney and are lost. We are getting in the shape of steam about 66 per cent. of the whole power in the coal, and losing about 34 per cent.

From this we see that the bulk of the loss of the fuel does not take place in the boiler; only a loss of 34 per cent. in the above case. In the case of an engine in which the horse-power represents one-eleventh of the whole coal burnt, the total waste of the whole engine, condenser, and boilers amounts to 91 per cent.. So that the boiler can be credited with a waste of only about one-third of the total waste.

Until a student is quite familiar with such calculations as the above, he must only look upon them as safe general guides to investigation into the causes of the waste of fuel; he must not consider them as absolutely correct. (I am not here speaking of the tables.)

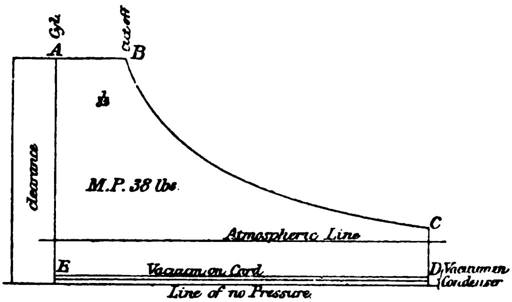
No engine and boiler ever probably worked exactly as we are describing this one. In one case you have a good boiler and a bad engine, and in another a bad boiler and a good engine, and most frequently both are bad. Then the rate at which the boiler is fired and a host of other considerations put all possibility of absolute correctness out of the question. When the student is familiar with them he will see where such calculations are liable to lead him astray.

From this point we may proceed to determine the power we are getting out of the pound weight of steam.

We have already assumed the pound of anthracite to be capable of evaporating 8 lbs. of water at a boiler pressure of 60 lbs. This pressure corresponds to a temperature of 307° Fahr., and at this pressure and temperature the steam occupies about 348 times the volume of the water from which it was formed, so that the 8 lbs. of water will occupy approximately $8 \times 348 \times 27 \cdot 7$ cubic inches = 77,000 cubic inches (1 lb. of water occupies $27 \cdot 7$ cubic inches).

We will now suppose these 77,000 cubic inches to be expanded in a cylinder, giving what is called an expansion of 6 times, taking $\frac{1}{10}$th clearance into consideration, and supposing there to be no loss from condensation, leakage, wire drawing, or cushioning, and that the expansion takes place according to Boyle and Mariotte's law. Such steam in expanding would give the indicator card in Fig. 50, and a mean pressure of 38 lbs. If the reader takes the trouble to

FIG. 50.



make the calculation he will find that a cylinder 63 inches diameter, with an 8-foot stroke, and $\frac{1}{10}$th of the stroke clearance, would require 77,000 cubic inches steam to fill it with boiler pressure steam, and give an expansion of 6 times.

The foot-pounds done by this steam in expanding are found by multiplying the area of the cylinder in inches, by the stroke, and by the mean pressure in lbs., viz. 38. This gives 940,000 foot-pounds; but, according to Joule's experiments, 772 foot-pounds represent 1 unit of heat Fahr.

We have therefore transformed into work $\frac{940000}{772}$, or 1220 units of heat. Let us compare this quantity with the total heat in the steam. The steam contained 8600 units, and out of these we have utilized 1220 units, or $\frac{1}{7}$th approximately. This $\frac{1}{7}$th is $\frac{1}{7}$th of the heat in the steam, not $\frac{1}{7}$th of the heat in the coal. The heat in the steam was $\frac{66}{100}$ of the heat in the coal, so we have developed as foot-pounds $\frac{1}{7}$th of $\frac{66}{100}$ of the heat in the coal, or $\frac{1}{10 \cdot 6}$.

From the cylinder the steam passes into the condenser, from which it is returned to the boiler as feed water at 132° Fahr., according to supposition. And the whole of the heat in the steam which we put into it, after deducting the 1220 units which appear

in the cylinder, viz. 8600 − 1220 = 7380 units, passes away in the circulating water, and is irretrievably lost. It must not be forgotten that throughout these calculations we have assumed the feed water to be at a temperature of 132° Fahr.; but if the feed had been at a higher temperature the evaporation per pound of coal would have been higher, not very much, but still it would have been so to a certain extent, and the horse-power per pound of fuel would have been greater. With a feed at 132°, and a so-called expansion of 6 times, we see that 7380 units of heat in the steam are lost out of a total quantity of 8600, or very nearly 86 per cent.: 14 per cent. of the heat of the steam which leaves the boiler is used, 86 per cent. is thrown away.

We see from this that the great loss of fuel at present is not in the boiler, but lies between the cylinder and the condenser: the boiler does 66 per cent. of what it should do, the engine only does 14 per cent. of its duty. And what is still more to the point is that not only the relative inefficiency of the engine is greater than that of the boiler, but it is so absolutely; for we find, by referring to page 74, that out of all the heat developed by 1 lb. of anthracite, 4400 units go up the chimney and are wasted by the boiler, whereas out of all the heat generated by 1 lb. of anthracite no less than 7380 are thrown away in the circulating water, or 1·7 times as much coal is wasted absolutely by the engine as is wasted by the boiler.

The effect of a warm feed water is to economize fuel. The boiler evaporates more water per pound of fuel, because to turn the feed water into steam it begins higher up the scale of temperature, but the saving is due to the condenser, which does not allow so much heat to pass away in the circulating water. In the example we have been considering 1 lb. of anthracite gave us 8 lbs. of steam, and put into them 8600 units of heat, but if the feed water had been at 140° instead of 100° above the freezing point, we should have saved 40° units on each lb. of feed water, or a total of 320° units; and instead of getting steam containing 8600 units we should have got steam containing 8920 units, or nearly 4 per cent. more power.

CHAPTER IV.

CONSUMPTION OF STEAM.

In the last chapter, the fact that there is a great waste of heat in the engine and condenser was pointed out. It now behoves us to go more into the details of the question, and to endeavour to discover what are the causes of loss, and how much of the loss is to be assigned to each particular cause.

Various authorities have given us their views on these questions, and it is not a little remarkable that there should be so much apparent difference of opinion among them. It is necessary, in the consideration of their views, to bear in mind that the engine of 1876 is a very different thing to what it was ten or fifteen years ago, and that what was an absolutely correct opinion when applied to the older engine, is not necessarily so when applied to the more recent one.

I make this remark because I shall presently quote the observations and opinions of undeniable authorities who are now no longer living, and because some of their views are not at all in accordance with the best opinions of the present day. Their views are, nevertheless, of great value to us, not only in illustrating the progressive development of the engine, but as a caution against forming general conclusions too readily, even when such conclusions are based on a very extended experience.

Economy is generally supposed to be attained by a considerable degree of expansion, but if we turn to vol. xvi.* we shall see some very curious remarks on this subject by the late Mr. Humphrys, a man who was as well qualified to speak as any living. He said: "In fact he had never found an instance in a steamboat where the cost of the power was less, when working expansively, than it was when working full power. The moment a marine engine began to work expansively, beyond what was due to the set and lap of the

* 'Proc. Inst. Civ. Eng.,' p. 362. 1856–7.

slide, the speed of the piston was reduced, and an important element in obtaining economy in working steam expansively was lost."

And he gave the following table for H.M.S. 'Fury,' a paddle-steamer 515 horse-power. They are the records of her performances for three or four years. The calculations were made by Mr. Humphrys from data supplied by the engineer :

	Coal Consumed		Distance run per Day in Statute Miles.	Distance the Coals would last.
	Per Day.	Per I.H.P.		
Full power	50	3·97	264	2112
1st grade expansion	40	4·24	240	2400
2nd " "	36	4·79	222	2466
3rd " "	33	5·46	211	2555
4th " "	28	5·57	190	2698
5th " "	22	5·8	155	2790
6th " "	19	7·53	130	2730
7th " "	15	8·87	110	2926

The late Mr. Dinnan said that he had found on H.M.S.S.'Dee,' a paddle engine working at 10 lbs. pressure, that when the steam was cut off earlier than half stroke the consumption of fuel was increased immediately.*

This question of expansion has been discussed continually by engineers, and it is one which will, most probably, continue to be so for long; but besides this there are many others, among them the one, Is a high boiler pressure economical ?

Mr. T. R. Crampton, speaking to Mr. Bramwell's paper on marine engines,† gave some interesting results of experiments conducted by Mr. Humphrys and himself thirty years ago, on a single-cylinder pumping engine, jacketed, 18 inches diameter, 20 inches stroke; there was an ordinary slide with expansion slide on the back, and a surface condenser. The first experiment was with steam at 70 lbs. pressure above the atmosphere, and an expansion of 12 times. The pressure was reduced 10 lbs. at a time in successive weeks, until it reached 35, with an expansion of 6 times. The consumption of fuel compared to the water lifted remained practically the same, and amounted to $2\frac{3}{4}$ lbs. per indicated horse-

* 'Proc. Inst. Civ. Eng.,' vol. xviii. pp. 266–7.

† 'Proc. Inst. Mec. Eng.,' 1872, p. 166.

power per hour when the indicated horse-power was measured by the water actually raised, and $2\frac{1}{4}$ if measured by the indicator. From this it might be safely inferred that, in this particular instance, high boiler pressure, with an early cut off, was not more economical than a low pressure and late cut off.

Then there are such questions as the following, which an engineer has to answer: What the effects of expansion in two or more cylinders are, of high piston speed, surface condensation, wire drawing, jacketing, slow combustion, &c.

Most of these questions will be referred to in this chapter; but it is not pretended that anything like all that can be said, about questions on which such variety of opinion exists, has been said, or is even known to the author.

The economy in point of fuel of an engine, especially a condensing one, is dependent upon so many conditions, varying independently, that it is extremely difficult to say what is, or is not, the cause of any saving. If an engine be tried most carefully under a certain set of circumstances, and then again tried, one of those circumstances only being changed (to all appearances), as has been repeatedly done, the difference in the results cannot be with certainty attributed to that solitary change; for, although the engine works apparently under precisely the same set of circumstances, one only excepted, this is not really the case, for it is perfectly impossible to change one circumstance without changing one or more of the others.

As an example, I will refer to Messrs. Crampton and Humphrys' experiments above. They found that increased boiler pressure gave no more work for a pound of steam condensed than the lower pressure gave; and assumed high pressures to be not economical. But supposing we consider that higher pressure in the boiler must mean a wider limit of temperature through which the metal forming the cylinder has to oscillate, and reflect that the wider this limit the less economically the steam must work, we see that if precautions could have been taken to prevent this an economy might probably have been formed in working steam at a high pressure. And again when the pressure fell, in the experiments, the piston speed very probably fell also. There is however so little doubt, judging by more recent experience, that

high pressure is a most efficient means of economizing steam, that nothing more need be said on the subject here.

Mr. James Wright has given * the following interesting table, showing the results of a trial which was made in July, 1865, on the service boiler of H.M.S. 'Oberon.' The pressure must be ascertained.

Water evaporated to 1 lb. of Coal consumed.	Coals burnt per Square Foot of Fire Grate per hour.	Total Coal used.
9·2	15·57	6,884
8·6	17·69	7,676
8·43	24·04	10,594

The temperature of the feed was close upon 65° Fahr. Welsh coal was used; the total fire bar was 75·98 square feet; the trials lasted close upon $5\frac{3}{4}$ hours. Comparing the first with the third trial, we see that although the coals were burned in the third case more rapidly than the first in the proportion of 1 to 1·54, the evaporation has only fallen off in the ratio of 1 to ·92. So that although we lose 8 per cent. more of the fuel, we get half as much more work in the same time out of the same boiler shell.

In the case of the United States revenue steamer 'Rush,' the evaporation per lb. of fuel at 69 lbs. pressure was 7·55 lbs. of water, and the rate of combustion per square foot of fire bar per hour 11·4 lbs. The results in this case were of a most reliable nature. The experiments will be referred to again at full length on p. 127. The trial lasted fifty-five hours.

We now come to the question of superheating. There is abundant testimony that coal is economized by superheating, but there is an obvious limit to the desirable amount of it, which is the temperature. If this rises too high we begin to destroy all lubrication, faces begin to score, and packing begins to burn, and we find that in compound engines working with steam at temperatures corresponding to pressures of 60 to 80 lbs., superheating has not been adopted to anything like the extent it was in the days of 20 to 30 lb. steam. Although we are approaching this limit when we

* 'Report of the Committee on Designs,' p. 210.

work with such steam, it does not appear that we have reached it, for we have locomotives working at 140 lbs. pressure and running at piston speeds of 800 or 900 feet, and in the case of some London and North-Western engines by Mr. F. W. Webb, even 1200 feet a second, which is considerably in excess of marine speeds.

Mr. Dinnen, when speaking of H.M.S.S. 'Dee,' whose engines, with jacketed cylinders, were made in 1832 by Messrs. Maudslay and Field, said her "best performance when working at full power, with common steam over a period of ten hours, was 3·9 lbs. of Welsh coal per indicated horse-power per hour." But "when the steam after leaving the boilers at a temperature of 237°" (corresponding to about 9 lbs. per square inch of boiler pressure) "had been superheated to 377°, the result was 2·74 lbs. per indicated horse-power per hour taken over a period of ten hours, as in the experiment with common steam. The expansion gear was employed during this trial, the steam being cut off exactly at half stroke."

The late Mr. Humphrys, speaking * in the year 1860, made the following remarks upon the performance of the Peninsular and Oriental Steamship Company's vessel 'Ceylon': firstly, without superheated steam; and secondly, with it. The cylinders were not jacketed, 72 in. diameter, 3 feet stroke, going from 50 to 60 revolutions. On an average of seven passages with ordinary steam the consumption per voyage there and back was 1500 tons. He then applied a superheater, which increased the temperature of the steam rather more than 100°. She had since completed five voyages, and her average consumption was 1100 tons; the cylinders, slides, &c., were in better condition than before. The saving here was 400 tons on 1500, or close upon 27 per cent. of the amount consumed before superheating.

In a ship named the 'Alhambra' the consumption fell from 19 to 13 cwt. per hour. And in the screw steamer 'Nepaul,' from 36 to 17 tons a day, which is a very substantial saving.

On the same occasion the late Mr. Beardmore, of the General Steam Navigation Company, mentioned the case of a vessel running between the Thames and Scotland in which on an average of twelve voyages the consumption had been 126 tons previous to super-

* 'Proc. Inst. Civ. Eng.,' vol. xix. p. 473.

heating, and which fell to 90 tons when the steam was superheated 100°.

How it happens that the addition of the small quantity of heat to the steam which is added by superheating enables us to get so much more power out of it will be considered later on. That the quantity of heat added is small, we shall see at once.

The case of the 'Dee' may as well be taken as any other to show this.

With plain steam it was found that per indicated horse-power the consumption was 3·9 lbs., and supposing the evaporation was 7 lbs. of water, then the water per indicated horse-power per hour was $7 \times 3 \cdot 9 = 27 \cdot 3$, which I shall call 27 lbs. of steam at a pressure of 9 lbs., and a corresponding temperature of 237° Fahr. One pound of such steam contains 1154 units of heat, and 27 therefore contain 31,200 units, and for this expenditure of steam we get 1 indicated horse-power per hour.

Now, taking superheated steam, we got an indicated horse-power for 2·7 lbs., and supposing the firing the same in both cases we get $2 \cdot 7 \times 7 = 18 \cdot 9$, say 19 lbs. of steam, these contain before superheating $19 \times 1154 =$ nearly 22,000, and after superheating do as much work as the 31,200 units above.

Now let us see how much heat we have put in. By superheating we have raised the temperature from 237 to 377, viz. 140° Fahr., and the weight of steam so raised is 19 lbs., so that if we had been heating water we should have put in $19 \times 140 = 2600$ units of heat; but to raise a pound of steam through 1° Fahr. only requires approximately half of the amount of heat necessary to raise 1 lb. of water,* so that when we raised the 19 lbs. of steam through 140° we put into it $\frac{19 \times 140}{2} = 1330$ units.

Now going back, we had 22,000 units in the steam before superheating, and then put 1330 more in, giving us a total of 23,330, and out of this we get as much work as out of 31,200 units of heat in saturated steam. So that it must be seen clearly enough that the work done by superheating steam is not proportioned to the amount of heat which we put into it when we superheat it, and further, that the total quantity of heat in the

* Regnault.

steam is only one of the circumstances which determines how much work we shall get out of it. This seems to depend upon the way in which the heat is contained in the steam, as well as upon the amount of it.

It is probable that with unjacketed cylinders the results would probably have been less remarkable. To this subject reference will be made when speaking of the advantages of jacketing, with which it is very closely associated.

We shall now go on to consider the question of the expansion of the steam in the cylinder, together with the important and closely associated question of the exhaust of the steam into the air or condenser, as the case may be.

If we take an ordinary unjacketed cylinder and admit into it steam at 30 lbs. pressure from a boiler, the first effect will be that a large amount of the steam will condense upon coming in contact with the cold cylinder, cylinder cover, and piston. Gradually the temperature will rise until we shall eventually have in the cylinder the full boiler pressure of 30 lbs., a large amount of condensed water at a temperature corresponding to a pressure of 30 lbs., viz. 274° Fahr., and the cylinder, cover, and piston will also be at this temperature.

We will suppose in the first place that we have the whole cylinder full of steam at boiler pressure, that is to say, that we don't expand the steam at all. Now suppose the slide valve to open the exhaust port, so that the steam can rush out into the open air, as in the case of an ordinary non-condensing engine. Of course the steam, being at a pressure of 30 lbs. above the atmosphere, will rush out violently at first, and less and less so as the pressure is relieved by the escape. The steam will, however, continue to go out until the pressure in the inside of the cylinder has fallen until it is equal to the pressure of the air on the outside.

Then matters would come to rest provided there was no condensed water in the cylinder. But there is a large quantity there, and it is all at a temperature of 274°, and the sides and ends of the cylinder, cylinder cover, and piston are at this temperature also. The consequence is that as soon as the steam begins to escape through the exhaust port, and the pressure to fall, this condensed water begins to boil, for it is at too high temperature to remain as

a liquid at this diminished pressure. It goes on boiling, and the hot cylinder with which it is in contact gives up some of its heat to the water and keeps it at the boil, for water at a temperature of 274° does not contain sufficient heat to evaporate itself: counting from the freezing point, the water contains only 242 units of heat, but saturated steam at 274° contains 1166 units, if therefore all the water were evaporated at a pressure corresponding to 274°, 924 units would be abstracted from the cylinder,* but as the pressure keeps falling, consequently somewhat less than 924 units will be required, and the result is that the temperature in the cylinder, by the time the exhaust port closes, is considerably below the temperature corresponding to a pressure of 30 lbs. Now we will close the exhaust and admit a fresh rush of steam from the boiler. The hot 30 lbs. steam again meets with a cylinder, cylinder cover, and piston at a temperature considerably below its own, and therefore we have again a condensation of steam, and when the exhaust port opens we shall have precisely the same evaporation going on again: so that every time steam is admitted into the cylinder a certain quantity of it is condensed at once. If we had used a condenser this condensed steam would have increased in amount for the reason that when the exhaust port opened the steam would escape more easily, and the boiling off of the condensed steam, or, as it is commonly spoken of, the condensed water, would go on more rapidly, therefore more would evaporate, and therefore more heat would be carried out of the cylinder each time.

Now precisely the same effect would be observed, but even in a more marked manner, if instead of using steam all through the stroke we had used it expansively. This cooling effect of the exhaust, and especially in engines working with high grades of expansion, is one of the serious causes of the loss of steam, and this loss becomes greater as the degree of expansion increases, as compared with the losses from other causes.

In the case where we use the steam expansively the re-evaporation of the condensed water begins as soon as the pressure begins to fall owing to expansion, in fact directly after the cut off, and does not wait for the exhaust port to open, as it did in the first case we considered.

* For each pound of water evaporated.

And here we see one reason why the compound engine should tend to give us an economical result: the steam from the high-pressure cylinder passes out of it into the receiver without undergoing any great amount of expansion, thus (approximately) if the low-pressure cylinder cuts off at half, and is three times the area of the high-pressure cylinder, the expansion into the receiver will be from a volume 1, to 1·5 at the most, and consequently the high-pressure cylinder is not cooled to anything like the extent to which it would have been if the steam had had a free rush into the condenser. But besides the condensed water produced by the cooled metal of the engine, it has been said that there is also a certain amount produced by a very different cause, which is the actual expansion of the steam.

We shall see later on in the chapter on the expansion of gases, page 123, that if saturated steam be allowed to expand, doing the full amount of work as it expands, then partial condensation will take place. But whether the case of steam expanding in a cylinder and the one supposed are analogous, is not a fact so easily proved. The author's impression is that the cases are not so.

Exactly the same condensation takes place when we see the cloud of white exhaust steam from any engine. The same holds good in the case of air and steam mixed, as may be proved by observing what takes place in the bell jar of an air pump. Air contains a certain amount of steam or aqueous vapour, and after a stroke or two of the air-pump plunger, a white cloud is produced by its condensation.

That which destroys the analogy between steam expanding as it ordinarily does in a cylinder, and steam expanding as in the case supposed above, is that there is something else in the cylinder besides the steam, namely, hot water, and this hot water by boiling off again may, I don't say does, but may, prevent the steam from condensing. But it must not be forgotten that even if there is an actual condensation in the expanding steam, such a condensation brought about not by any loss of heat from radiation or convection, but simply by the performance of work, would be an economy and not a loss, inasmuch as for a small quantity of condensed steam there would be given in exchange a large amount of sensible heat, which would go to increase the pressure of the steam

in a far greater extent than the pressure would fall off due to the condensation of a certain portion of it.

Thus, if $\frac{1}{5}$ of the one pound weight of steam at 45 lbs. absolute pressure were to be condensed, we should diminish the volume to $\frac{4}{5}$ of what it was, and liberate $\frac{1}{5}$ of the latent heat of steam at 45 lbs. which is 923, we should liberate 185 units. Now, assuming the specific heat of steam to be $\cdot 5$, we shall raise the temperature of what steam remains as a vapour $\frac{\frac{5}{4} \times 185}{\cdot 5}$ = about 461° Fahr., and our pressure will be

$$\frac{459 + 275 + 461}{459 + 275} \times \frac{4}{5} \times 45 = 58 \cdot 6 \text{ lbs. (see p. 107).*}$$

so that although we have only $\frac{4}{5}$ of the weight we started with, we have kept the volume the same, and the pressure is 59 as compared with 45 which we had at first.†

There is another way of looking at the fact that condensation produced by work done during expansion is no loss. The amount of heat which goes into the condenser plus the heat equivalent to the work done while expanding, is equal to the total heat in the steam when first admitted. Now, if any of the steam be condensed in the cylinder, then the mixture of steam and water which blows itself into the condenser at the end of the stroke, contains less heat than if it had all been steam, therefore more heat must have disappeared as work.

It would almost at first sight appear as if when a cylinder was jacketed, we were preventing an advantageous condensation, and merely warmed the steam to blow it when so warmed straight into the condenser, getting a little extra power due to the steam being slightly warmer.

Jacketing is not resorted to for the purpose of warming the steam; if any engineers have jacketed cylinders with that view the author ventures to think they have been mistaken. The jacketed surface exposed to the steam is so small, and the time the steam

* 275° Fahr. correspond to a tension of 45 lbs., and $\frac{4}{5}$ is introduced on account of the increased volume.

† This could not of course occur, it is merely an example showing the tendency of condensation.

is in contact with it so short, that we should have been led at once to suppose that only a very slight transference of heat from one to the other could take place. Steam, moreover, is, like all other gases and vapours, a very poor conductor of heat.

But we have independent proof of the fact in the engines of H.M. Steamship 'Briton,' which was fitted with a tubular superheater between the high and the low pressure cylinders, heated by steam at boiler pressure direct from the boilers in the same way as an ordinary steam jacket. Messrs. J. and G. Rennie, the makers of those engines, have kindly given me the following particulars of the engines: diameter high-pressure cylinder, 57 inches; diameter low-pressure cylinder, 100¼ inches; stroke, 2 feet 9 inches; revolutions, 95½; boiler pressure, 58 lbs.; square feet of heating surface in the above superheater, 114;* cut off, 21½ inches in the high-pressure cylinder. From this we may easily calculate that 6061 cubic feet of steam were passing through the superheater per minute, and also the number of feet of heating surface in the high and low pressure jackets together, and cylinder covers and ends which amounts to about 185 feet if both cylinders had been jacketed.†

Mr. Wright, of the Admiralty, in his evidence before the Committee on designs, said that the difference in temperature between the steam when it went into the receiver and when it came out was only 2°; and that with the steam in the jacket of the receiver (by which was meant the superheater spoken of above), the steam came out at 1° higher temperature than it went in; without the steam in the jacket it came out apparently at 1° less temperature than it went in.

Now to deal firstly with the relative heating surfaces of the superheater, and of the combined cylinders. The superheater had 114 square feet in it; the two cylinders, covers, and ends had 185 square feet in them. The ratio is 1 to 1·62.

During more than half the time the steam in the low-pressure jacket was not heating working steam, but was heating the cylinder metal, which in its turn was heating the exhausting or waste steam; again the superheater was not only made of a much

* I calculated this area from a sketch sent me by Mr. G. B. Rennie.

† The high-pressure cylinder was not jacketed.

better conductor, viz. brass, which is more than twice as good as iron, but the material through which the heat had to pass was not $\frac{1}{3}$ of the thickness of the cylinder metal; and lastly, the steam in contact with the superheater was very dense compared with what it subsequently became in the cylinders, and was, moreover, brought into a very much better contact with the heating surface flowing over it in a thin film, instead of merely lying against it as in the cylinder, so that the superheater was working at an immense advantage compared to the jackets.

If, then, the superheater could only raise the temperature of the steam 2°, it is hardly to be expected that the jackets, although presenting a little more surface, should be able to raise the temperature anything like as much as 2°.

From the action of an ordinary superheater in the chimney, we may draw the conclusion that the heating surface of the jacket is not sufficient to raise the temperature of the steam to any appreciable extent. Take a large engine with a jacket heating surface of 520 square feet. The superheating surface in the chimney was 2260 square feet, that is, 4·3 times as much as in the jackets, and was of an immensly better description than that of the jackets, for, firstly, the steam was in motion through its tubes instead of being merely in contact, and, secondly, the metal of the superheater tubes is very thin as compared with that through which the jacket heat has to get, and, thirdly, the difference of temperatures on opposite sides of those metals is many times greater in the superheater than in the jacket.

The steam in passing through the superheater, which was a wrought-iron tubular one, was sometimes not superheated more than a few degrees above the temperature corresponding to the pressure in the steam pipe, and never more than 30° Fahr. If, therefore, it had passed through only $\frac{1}{4 \cdot 3}$ of a much inferior surface, the temperature could not have been raised much.

But again we may look at this question from another point of view. Mr. I. Lowthian Bell has most obligingly given me the following information relating to his blast heating stoves: Each stove contains 1700 square feet of heating surface, and, if the pump delivers its full displacement, heats 1617 cubic feet of

air per minute to 1075° Fahr. Now I have assumed the pump to deliver 80 per cent. of its displacement, and find that for every pound of air passing through the stove per minute there is an allowance of 16·2 square feet of heating surface. Mr. Rees, of the Etruria Furnaces, has kindly given me the particulars of a large pair of blowing engines supplying three blast furnaces.* The blowing cylinders, two in number, are 100 inches diameter, and 9 feet stroke; and I find, supposing the pumps to deliver 80 per cent. of their displacement, that for every pound of air passing through the heating stove per minute, there is an allowance of 18 square feet of heating surface, and the temperature is raised from say 100° up to 1000°, or a rise of 900° Fahr.

Now let us compare this with the square feet of jacketing surface per pound weight of steam passing through the 'Briton's' engines, assuming both cylinders to have been jacketed. If we consider that 6061 cubic feet of steam at 55 lbs. pressure passed through the 'Briton's' engine per minute, that would amount to 1000 lbs. If, taking blast-furnace proportions, we had wished to superheat to the extent of 900°, and remembering the specific heat of steam to be, according to Regnault, double (approximately) that of air, we ought to have had a surface of $1000 \times 18 \times 2 = 36000$ square feet. But we had in the jackets only 185 feet, assuming both cylinders jacketed, and therefore, had the quality of the heating surfaces been the same, we might have expected a rise of temperature of $\frac{185}{36000} 900 = 4 \cdot 6°$ Fahr. But the heating surfaces were very far from being of the same quality, for in one case the heat was coming from boiler steam of a temperature of 300° Fahr., in the other the heat came direct from the waste products of a fire, and may have been anything between 1500 and 2500° Fahr. This comparison would, therefore, lead us to suppose that the jackets can do very little in raising the temperature of the circulating steam. It may be here noticed that the amount of heat which will travel through a conductor from one side to the other is very dependent upon the temperatures of the two sides of the conductor: it is proportional to

* There were two combined engines. Revolutions per minute, 9. Equal to 36 single strokes. Furnaces supplied, 3. Stroke, 9 feet. Diam. blast cyl., 100 in. Heating surface per furnace, 6794 square feet.

the difference between those two temperatures. In the case of the jackets this difference of temperature is very small, at one time so small as to be practically nothing, just when the cylinder is taking steam, and can never amount to much more than 100° Fahr., perhaps the average is much less than this. But in the case of the blast furnace the difference was most probably never less than 500°, and may have been 2500° or more. When the difference was greatest in the jackets it was 100° Fahr., when greatest with the furnace it may have been 2500°.

If the advantage of jacketing does not lie in the heating of the steam, where does it lie? We have seen that the cylinder is inevitably cooled by each operation of exhausting, and it appears to the author that the whole question turns on this point, is it better to let the working steam re-warm the cylinder, or is it better to warm the cylinder by jacketing? This we shall see best by taking an example. We shall see it at once if we take an extreme one, and suppose $\frac{9}{10}$ of all our admitted steam to be condensed, and only $\frac{1}{10}$ to remain as vapour or steam. Now, if we compare the amount of steam by weight which is to do work by expanding with the total amount which is taken out of the boiler at each stroke, it is approximately 1 to 10 (not exactly, because of subsequent re-evaporation). Suppose now we had jacketed the cylinder so that none of the steam on going into the cylinder were condensed, the proportion is a much more favourable one, for we have approximately 9 lbs. of steam condensed in the jacket, and 10 taken into the cylinder, total 19, of which 10 do useful work in expanding; the proportion is then 1 to 1·9 instead of 1 to 10 as in the first case. Such a calculation as the above does not amount to a proof, but merely shows the manner in which jacketing may probably be beneficial.

Suppose we take 1 lb. of steam at about 60 lbs. pressure, that is, at about 45 lbs. boiler pressure, and enclose it in cylinder of such a diameter that when the piston has just moved 12 inches, it will exactly hold 1 lb. of steam. If this steam were to expand, following Boyle's law, we should get a mean pressure of 24·, supposing there to be an expansion of 5 times, and the work done is the area of the piston $\times$ 24· $\times$ 4 feet + the area of the piston $\times$ 60 $\times$ 1 foot = area $\times$ 156 foot-pounds. This is approximately

the work we should get out of the steam if there were no condensation. We now take into consideration that there is condensation, and suppose 15 per cent. by weight of the steam to be condensed.

We then have left in the shape of steam $\frac{85}{100}$ of the original quantity, but occupying the same volume. We must first see what its specific volume is, and then, by consulting the tables, find the temperature and pressure corresponding. The specific volume of steam at 60 lbs. tension is about 425, if $\frac{85}{100}$ occupy the volume which one should, the specific volume becomes $\frac{100}{85} \times 425 = 500$. Looking at the table we learn that a pressure of $50 \cdot 6$ lbs. and a temperature of 282° correspond to a specific volume 500. The work we should get out of this steam by expanding is the area of the piston $\times\ 50 \cdot 6 \times 1$ foot $+$ the area multiplied by the mean pressure during expansion, or $20 \cdot 25 \times 4$ feet $=$ area $\times\ 131 \cdot 6$ foot-pounds, and the work done without condensation was area $\times$ 156 foot-pounds. Therefore we have done $24 \cdot 4 \times$ area, less units of work.

Now, having compared the amount of work done by the two quantities of steam, let us compare the total quantities of heat in each. At a pressure of 60 lbs. a pound weight of steam contains 1171 units of heat; at a pressure of $50 \cdot 6$, 85 per cent. of a pound contains 1000 units and $\frac{15}{100}$ of a pound of water at a temperature of 282° contain 37 units, so that the steam and water together contain 1037 units. We have therefore taken out $1172 - 1037$ units $= 135$ units. We have abstracted $\frac{135}{1172}$ or $\frac{1}{9 \cdot 4}$ of the heat, and lost $\frac{24 \cdot 4}{156}$ or $\frac{1}{6\frac{1}{2}}$ of the work.

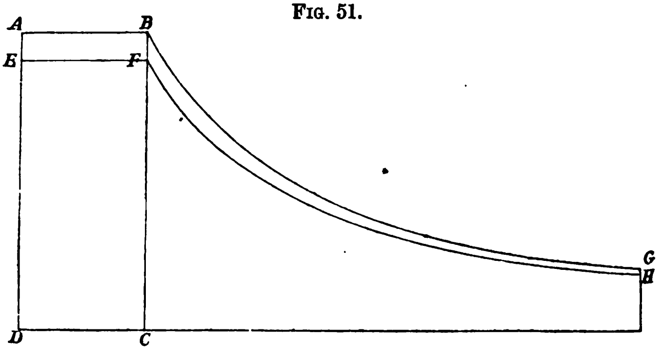
I just give the results of the effect of condensing $\frac{1}{4}$ of steam at 100 lbs. boiler pressure or 115 lbs. absolute, expanding 5 times. We shall then find we have lost 27 per cent. of the power, and taken away $18 \cdot 6$ per cent. of the heat. And the higher in pressure we go, we shall find that the ratio of the power lost compared to the heat abstracted, becomes greater.

The value of jacketing then consists in supplying this heat abstracted by the exhaust, in a more economical manner than from

the working steam, and thereby by saving that heat which should do work in the steam. By jacketing we do not prevent the condenser from abstracting the heat from the cylinder, but we prevent the cylinder from re-heating itself in a wasteful manner, and we prevent the temperature of the cylinder from oscillating through wide limits.

There is a graphical method of considering the effect of this saving of heat. Let, in Fig. 51, the line A B G represent the diagram produced by the uncooled weight of steam expanding from an initial pressure of 60 lbs., and let the line E F H represent that produced by the cooled steam expanding from an initial pressure of 54 lbs. after a condensation of $\frac{1}{10}$. If by our jacketing we prevent the steam from condensing, we shall get the power due to the line A B G, that is, we shall gain power represented by the area of the figure A B G H E F, having merely put in an amount of heat represented by the figure A B F E.

FIG. 51.

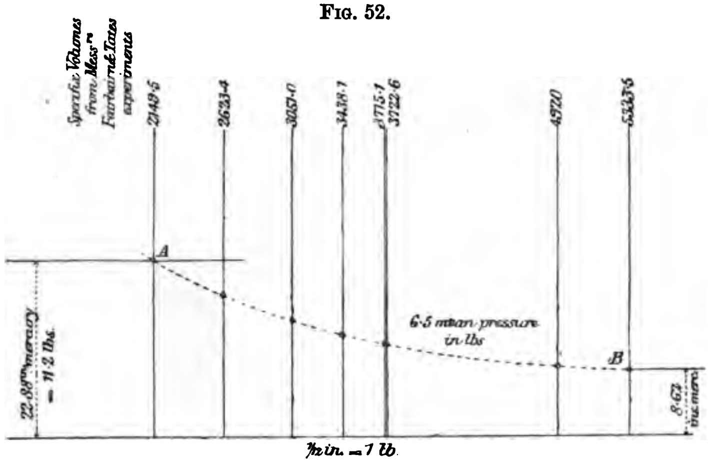


I am inclined to believe partly from the above considerations that economy would result from jacketing in a cylinder which did not work expansively, but that the more expansively the steam is used, the greater would be the economy in the use of the jacket.

In the previous considerations we have made an assumption that the steam follows Boyle's law in expanding, but this probably does not lead us into any great error, for we applied the same law to both cases. And again, we have assumed what never is the fact, viz. that one set of actions take place and finish before another set begin, whereas they are both going on together.

Thus the steam, as in Fig. 52, does not lose all its heat between A and B before the expansion begins, but only loses a part there, and the rest as the piston allows the steam to come into fresh contacts

FIG. 52.



with the cylinder by moving on, so that really all that has been said amounts to this, if steam and steam jackets acted in the manner assumed, the results would be something near what is indicated by the diagram: but each engineer must use his own knowledge in determining to what extent such a course of analogous reasoning is applicable to the steam engine as it is.

If we are prepared to admit that this cooling effect of the cylinder is the chief or a source of loss in the cylinder, we see at once why a high piston speed must tend to produce an economy. If the piston speed is made greater, we certainly make the friction required to turn the engine greater, and so we have a loss at once, but at the same time a more than compensating advantage in diminished cooling surfaces, from which evaporation can take place into the condenser. If D is the diameter of a cylinder and S the stroke, these surfaces are approximately $2 \times D^2 \times \cdot 785$ for the cover and piston, and $3 \cdot 14 \times D \times S$ for the sides of the cylinder. There is no necessity to make a general formula of it; suppose we take a 20-inch cylinder and 20-inch stroke, then the total cooling surface is $13 \cdot 1$ square feet, found as above. But if we double the piston speed, that is, keep the stroke the same and diminish the

cylinder diameter (so as to keep the same indicated horse-power), our cylinder diameter becomes $\frac{20}{\sqrt{2}} = 14 \cdot 15$ inches; our cooling surface becomes $8 \cdot 3$ square feet. And these two amounts of cooling surface remain open to the condenser for the same length of time.

It is probable that the bad effect of this cooling is not directly proportional to the amount of cooling surface exposed to the condenser, because, after all, it is the amount of water evaporated off which determines the amount of cooling; still it is no doubt very closely connected with it, as increased surface offers increased facilities for this water evaporating.

In every engine the temperature of the cylinder must vary between two limits, but it is impossible to say what these limits are, and indeed they are most probably different, at different parts of the cylinder. At the two ends the cylinder is heated up to the steam temperature, and then cooled by evaporating into the condenser; in the middle it is never heated quite so much and remains slightly longer open to the condenser. However, at all parts of the cylinder there is this ultimate heating and cooling going on; what we shall call an oscillation of the cylinder, metal, piston, &c., between two temperatures takes place.

The action of a jacket is to simply diminish this difference of temperature. It is quite certain that the jacket never entirely annihilates this difference, but still its tendency is in the direction of doing so. The more effective it is, the less the cylinder falls below the temperature of the incoming steam, and consequently the less of the working steam is necessary to warm the cylinder. The amount of working steam lost by cooling and the amount of this oscillation between the two extreme temperatures are very closely connected.

But if this cooling effect due to this oscillation of temperature through wide limits is the chief source of loss in a cylinder, we may see a reason closely connected with the one referred to above why a compound engine should work more economically than a simple expansive one, and why a high-pressure very expansive engine is not one from which as much economy could be had as might be expected from the high-pressure and expansion.

If we consider the case of two simple engines working with 30 lbs. and 60 lbs. of steam, the temperatures of the steam are 274° and 307° Fahr., a difference of 33°; and the lower limit will be dependent on the vacuum, and will, if exhaust takes place when the steam is at the same tension, be much more nearly the same than the higher limits. In the case of the high-pressure steam working expansively, we are expanding the steam in a cylinder oscillating through wider limits of temperature than the low-pressure steam. Then, in reference to compounds, I may quote Mr. Wright's evidence.*

When asked his opinion as to the cause of the economy which was said to be obtained in the compound engine, Mr. Wright said:

"The only cause that I can see is that the variation of the temperature in the cylinders is much less in working the steam in two cylinders than it is in one;" and again, "You would have a much greater difference between the condensation temperature and the maximum steam temperature with a single engine than with a compound engine."

I have taken a large number of compound engines, and I find that on their trials the average indicated horse-power in the high-pressure cylinder amounted to 394, and in the low-pressure cylinder to 459, making a total indicated horse-power of 853, which numbers are in the proportion of 463, 537, and 1000. So we see that on an average in those compound engines actually 46 per cent. of the power is exerted without the steam ever being brought into contact with the condenser; the limits of temperature through which it works being the temperatures corresponding to the initial steam pressure and the pressure in the receiver. The average initial tension of the steam 68·6 lbs. corresponding to a temperature of 301° Fahr., and the average lowest pressure, as shown on the exhaust line of the high-pressure cylinder card, was 18 lbs., corresponding to a temperature of 222°. So that we were getting in these engines 46 per cent. of the whole power, where the steam only ranged through a limit of 79° Fahr.

It is quite impossible to say what the limits of the temperature are through which the different parts of the cylinder itself oscillate. In want of such information, the only thing we can do is

* Report of the Committee on Designs, p. 179.

to consider what the limits of the temperature of the steam are, and this must be our guide for the temperature limits of the cylinder, which are in both cases determined by the steam, for the higher limit is close to the steam temperature.

If we assume for the engines considered above an average vacuum as shown by the low-pressure card of 26 inches = or nearly 12·75 lbs., we have a tension of vapour corresponding to 14·75 − 12·75 = 2 lbs. (if the barometer stood at 14·75), which has a corresponding temperature of 126° Fahr. So we have 54 per cent. of the work done by a compound engine, between the limits of temperature of 222° and 126°, an oscillation of 96° Fahr. An engine, with boiler pressure of 60 lbs., and one with pressure of 30 lbs. exhausting down to a 26-inch vacuum in the cylinders, give out their whole power between limits respectively of 307° and 126°, a difference of 181°, and 274° and 126°, a difference of 148° Fahr.

I cannot do better than quote the late Professor Rankine's views on this question.* He says:

"One of the earliest consequences deduced from the principles of thermodynamics was, that when steam performs work by expansion, a quantity of heat disappears, sufficient not only to lower the temperature of the steam to that corresponding to its lowered pressure, but to cause a certain portion of the steam to pass into the liquid state. The steam thus spontaneously liquefied collects in the form of water in the cylinder; and if the cylinder and piston were made of a non-conducting material, the water would simply be discharged from time to time into the condenser without causing any waste of heat. But the cylinder and piston being made of a conducting material, give out heat to the liquid water which adheres to them, so as to re-evaporate it when the communication with the condenser is opened; and that heat is carried off to the condenser with the exhaust steam, leaving the piston and the inside of the cylinder at a low temperature, even though the outside of the cylinder should be clothed with an absolute non-conductor. When steam from the boiler is admitted at the beginning of the next stroke, part of it is immediately liquefied through the expenditure of its heat in raising the piston and the inside of the cylinder

* Report of the Committee on Designs, p. 322.

again to a high temperature; the result being that at the end of the second stroke the quantity of liquid water which is re-evaporated and carries off heat to the condenser is greater than it was at the end of the first stroke. At each successive stroke that quantity augments, until it reaches a fixed amount, depending mainly on the difference of the temperature of the steam at the beginning and end of the expansion; and the effect is the same as if a certain quantity of steam at each stroke passed directly from the boiler to the condenser without performing work. In some experiments lately made the quantity of steam which thus ran to waste was found to be greater than that which performed work, so that the expenditure of steam was more than doubled.

"The remedy for this cause of loss is to prevent that spontaneous liquefaction of the steam during its expansive working, in which the process just described originates; and that is done either by enclosing the cylinder in a jacket or casing supplied with hot steam from the boiler, or by superheating the steam before its admission into the cylinder, or by both these means combined. The steam is thus kept in a nearly dry state, so as to be a bad conductor of heat; and the moisture which it contains, though sufficient to lubricate the piston, is not allowed to increase to such an extent as to carry away any appreciable quantity of heat from the metal of the cylinder and piston to the condenser."

The last paragraph, beginning with "The remedy for this cause," attributes the usefulness of the jacket to an action which the author has endeavoured to show does not take place to any appreciable extent, namely, the heating of the steam. The fine mist which would be formed during the expansive working of dry saturated steam would in all probability not settle on the metal at all, or certainly to a slight extent only; and it would be blown straight into the condenser when the exhaust port opened, and would consequently produce but a slight degree of cooling at the most. But the author has endeavoured to show that, after a certain amount of condensed water has once been formed, the probability is that no mist is formed during the expansion, owing to the re-evaporation of the already condensed water. The action which brings about the loss of steam in an expansive engine would be felt, but not to the same degree, even if no working expansion

ever took place in the cylinder. This action is the re-evaporation of water while the exhaust port is opened; and the good of the jacket (in the author's opinion) is to provide the heat necessary for such evaporation in an economical manner. It would be a great saving to carry the condensed water mechanically from the cylinder to the condenser, but the difficulty is that the cylinder *will* re-evaporate it.

Although the quantity of condensed water, if the whole cylinder, &c., were at the temperature of the boiler, would, as Professor Rankine states, be first formed out of the expanding steam, nevertheless, after it has been formed, the expansion, while doing work on the piston, so far from being the cause of a fresh amount of liquefaction, is the cause of the re-evaporation of part of the water already liquefied. And the source from which fresh supplies of condensed water are obtained is the newly entering boiler steam, which is directly condensed on the sides of the cooled cylinder.

Returning to the subject of the limits of temperature between which the power is developed, in the trials of the compound engine the 'Rush,' jacketed cylinders, 24 and 38 inches diameter, the amount of water evaporated, accounted for by the high-pressure indicator card, was ·9323; therefore only ·068, or $\frac{1}{14 \cdot 7}$ of the whole water evaporated by the boiler was condensed in the high-pressure cylinder and the two jackets. The card from the low-pressure cylinder accounted for only ·7355 of all the water evaporated, but only ·9323 of the steam went into it as steam, so that the low-pressure cylinder card accounted for about $\frac{73}{93}$, or ·79 of the steam going into it, while the high-pressure cylinder accounted for ·9323.

It has been frequently said that the expansion which takes place in the receiver of a compound engine is the cause of loss.

Whether there is any loss from such an intermediate expansion in engines made of the materials now used, is, I think, doubtful. But if the cylinders could be made out of an absolutely non-conducting material, it is certain that with the same cylinder capacity more power would be developed without an intermediate expansion

than would be developed with such intermediate expansion, using the same weight of steam in both cases.

In treating this question of intermediate expansion, in the very first place it is desirable that the term expansion should be properly understood.

There may be quite different sorts of expansion in the case of a gas: the gas may expand, doing all the work it is capable of through that expansion, as it does in the case of an ordinary cylinder, or it may expand, doing literally no work at all, as in Dr. Joule's experiment (see p. 119), where the gas was allowed to expand into a vacuum, and with no change of temperature as the result.

There is, again, a whole series of intermediate expansions between these two limits, viz. doing all the work it can, and doing none. Let us call the vessel containing the compressed gas A, and the other one B. Suppose that there was not a perfect vacuum in B, but that there was some gas left in it; when the communication is opened the highly compressed gas in A will expand and compress that in B until the tension of the gas in both vessels is equal. If a thin piston had been placed between them, the compressed gas would have pushed this piston back until the pressure on each side of the piston should have become equal. By expansion, then, we should understand an increase of volume simply; and if any work has been done during the expansion, this may be mentioned as a separate fact.

Now what constitutes the measure of the work done by the compressed gas while expanding, when a small quantity of gas has been left in the other vessel? The work done is clearly the work required to compress that which was in the other vessel B. The work done by the gas in vessel A while expanding has nothing to do with what I may call the indicator card of the gas in A while expanding. If the vacuum in B had been almost perfect, the gas in A might double its volume and do almost no work at all. The true measure is the diagram produced by the compressed gas in B. So we see that the work done by a gas in expanding is not necessarily in all cases measured correctly by the indicator diagram taken during the expansion.

Now in the case of a compound engine the words expansion in

the receiver are frequently used, but so far as I am aware it is not distinctly stated what sort of expansion is meant, whether expansion doing all the work the steam is capable of, or only a part of it.

For the sake of simplicity we shall firstly suppose that the contents of the receiver are very large as compared with those of the cylinders, so that the high-pressure cylinder exhaust steam rushing in and out does not alter the pressure in the receiver appreciably.

At the moment when the high pressure cylinder exhausts into the receiver, the steam rushing out of that cylinder finds the receiver already full of steam at a less pressure than itself. (At this time the steam port of the low-pressure cylinder is, of course, closed.) The steam from the high-pressure cylinder now does work in compressing that which is already in the receiver, and from what has been said it will be obvious that the foot-pounds of work expended by the high-pressure exhaust steam in compressing the receiver steam must be exactly equal to the foot-pounds of work received by the receiver steam when so compressed. To give an illustration: Suppose that the high-pressure exhaust steam were to force back a spring instead of the receiver steam. If the spring were now allowed to expand it would give out exactly the same number of foot-pounds as were put into it to compress it, or, in other words, the power lost by the steam has been simply transferred to the spring.

This fixed quantity of steam in the receiver might be called stationary steam (that is to say, the same weight of steam similar in every way), for it always remains in the receiver as long as the working of the engine is not altered, and acts precisely as a spring would do, being alternately compressed and relaxed. And inasmuch as the power necessary to compress this steam is each time given out again when it expands, this stationary steam can be productive of no loss. The steam and water passing through the engine from the boiler to the condenser at each revolution might in contradistinction to the stationary steam be called the circulating steam.

Of course the high-pressure exhaust steam and the stationary receiver steam mix, but no power whatever is lost by mixing these steams, for although they have done work on each other, none has

been done externally to the receiver. All that has been done is to rearrange for a short time the manner in which the heat is distributed, taking some from one portion of the steam and giving it to the other.

Now suppose the low-pressure cylinder to take steam: at the instant of cut off by the low-pressure slide it will have taken just the same weight of steam and water as left the high-pressure cylinder, and will have left the stationary steam in the receiver just as it was before it was compressed by the high-pressure exhaust steam. And not only will the low-pressure cylinder at the instant of cut off contain the same weight of steam and water as exhausted out of the high-pressure cylinder, but this steam and water will, when the work done on the low-pressure piston up to the point of cut off is added, contain the same amount of heat as they did when they exhausted from the high-pressure cylinder.

Whatever heat has been lost by the circulating steam up to this point has disappeared in the form of work, and from this point up to the moment when the exhaust begins (supposing there to be no condensation on the sides) whatever further heat disappears will also disappear as work, so that there has been no loss in the sense of loss of heat without the corresponding gain in work done. If we have not got as much power out of the steam so expanded as out of steam expanded in a single cylinder, it is because the useful expansion has not been pushed to the same extent.

To find the ratio of the volumes of the steam when the exhaust begins in the low-pressure cylinder to that when the cut off takes place in the high-pressure cylinder, and to call that the ratio of expansion, is in a compound engine wrong, if the term expansion has the same meaning which is usually assigned to it in speaking of single-cylinder engines, for we have seen that part of the expansion takes place when the steam does not do its full work. The true measure of expansion is the ratio of expansion in the high-pressure cylinder multiplied by the ratio of expansion in the low-pressure cylinder, but any calculation of the power of an engine founded on this ratio will not be quite correct, because, while passing from the high-pressure cylinder into the receiver, the steam tends to slightly superheat itself.

CHAPTER V.

GASES AND VAPOURS.

I PROPOSE in the following pages to examine in an elementary manner some subjects relating to gases and vapours which are of interest to engineers.

The difference between a gas and a vapour is this; a vapour is a gas near its liquefying point, so that the difference is not one of composition, but of condition. Steam is a vapour, but the atmosphere is a mixture of gases, for it is impossible by any means now known either to compress the gases which form the air until they become liquid, or to cool them until they become liquid, nor can they be liquefied by both pressure and cold combined. Steam would very soon become liquid under either influence. Ammonia and carbonic acid are gases considerably removed from their points of liquefaction, but nevertheless they can be liquefied by pressure and the abstraction of heat; and when they have been made to approach these points they would be termed vapours.

It is necessary to note this difference between gases and vapours, because they behave differently under certain circumstances, as will be seen later on.

And firstly, we will endeavour to learn the relationship which exists between the pressure and the volume of a gas.

It was discovered by an Englishman named Boyle, and also by an Italian experimenter, Mariotte, that if a gas be taken (not steam, for reasons we shall see presently) and enclosed in any vessel fitted like a cylinder with a piston, where its pressure and volume could be accurately observed, that if the piston were forced in on the gas until the volume of the gas was only one-half what it was when the experiment began, the pressure gauge would show that the pressure was now double what it was to begin with; and again, that if the piston was forced still farther in until the volume

was only one-third of what it originally was, the pressure gauge would now show three times the original pressure: and the same relationship exists when the piston was pulled out instead of being forced in; that is to say, when the volume was made twice as great, the pressure was only half as much, and when it was three times as great the pressure was only one-third. The temperature of all the apparatus and the gases had to be kept the same throughout. Boyle therefore concluded that the pressure and the volume varied inversely.

If the pressure and the volume vary inversely, for example, if when you double the one you halve the other, it is clear that the two multiplied together must always be equal to a constant, i. e. unchanging quantity; and accordingly Boyle and Mariotte's law is usually expressed by $p \times v = c$ where p is the pressure, v the volume, and c the constant quantity.

c is known, and whatever change you make in either p or v will produce a change in the other, viz. v or p, which may be found from the above equation. Suppose the volume to be 10 cubic feet and the pressure 60 lbs., the $p \times v = 10 \times 60 = 600 = c$. Now let p become 33 lbs., then $v = \frac{600}{33} = 18 \cdot 2$ cubic feet.

This is a law which holds good with all gases under the following conditions, that they shall be taken at such a temperature and a pressure, that either or both together may be varied through wide limits without the gas approaching that point where it begins to condense into a liquid, and that the temperature of the gas shall be kept the same throughout the experiments. If we take a gas, for example, like carbonic acid, when we work with atmospheric pressures and temperatures we may make wide variations either way with both pressure and temperature, and never come near the liquefying point. But when we come to consider steam, we shall find that although in practice it does so happen that when it expands the pressure follows the above law, we shall at the same time find that the temperature varies very much. If we were to put steam through the same experiments, and keep the temperature the same all through them, we should find its behaviour quite different. We are in fact much too near

its point of liquefying. Suppose we take 2 cubic feet of steam at the temperature of 212° Fahr., that is, of boiling water, and have an internal pressure of 15 lbs. on the square inch, that is, just sufficient to balance the atmosphere, and then forcibly compress it until the volume was reduced to 1 cubic foot, we should find that by the time the temperature had fallen to 212° again we should only have the same pressure we had to start with, no more and no less, but a quantity of the steam would have turned to water; but if we had experimented on atmospheric air at these temperatures and pressures, we should have found double the pressure when the volume was reduced to 1 cubic foot.

This law of the pressure varying inversely as the volume does not apply to steam as we see in the same way as to gases far above their liquefying points; but nevertheless when steam does expand the pressure does vary approximately inversely as the volume, but, as mentioned before, the temperature varies also, and not only the temperature varies, but when steam expands, as in an unjacketed cylinder, some of it condenses also; so that steam when expanding departs from the law of $p\,v = c$ in two marked manners. If we prevented the temperature from altering and prevented any of the steam from condensing, we should find that the pressure would no longer be represented even approximately by the formula $p\,v = c$.

We now come to another point of importance connected with gases, which is the effect produced in them by changes of temperature. Experimentally it has been determined that all gases expand to very nearly the same extent when they are raised in temperature through the same number of degrees, all other circumstances being the same. If a cubic foot of oxygen, nitrogen, hydrogen, or air are taken at a temperature of the freezing point, or 32° Fahr., and heated to 33° Fahr., their volume will increase $\frac{1}{491 \cdot 2}$ of a cubic foot. If the temperature had been raised 5°, viz. to 37° Fahr., their volumes would have increased $\frac{5}{491}$, and if the temperature had been raised 100° the volume would have increased $\frac{100}{491}$ cubic foot, and therefore at a temperature of 132° Fahr. the volume would have become 1 cubic foot $+ \frac{100}{491}$, or $\frac{591}{491}$ cubic foot.

If we call t the number of degrees of temperature above the freezing point, and V_{32} the volume at 32° Fahr., then the volume,

which we will call V_T, at any new temperature which is, we will say, T degrees above 32° Fahr., will be

$$V_{32} + \frac{T}{491} V_{32}, \quad \text{or} \quad V_{32} \left\{1 + \frac{T}{491}\right\} = V_T.$$

From this equation we can easily determine what the volume of the gas must be at 32° when we know what it is at any other temperature, by merely transposing the factors of the equation, thus

$$V_{32} \left(1 + \frac{T}{491}\right) = V_T,$$

therefore

$$V_{32} = \frac{V_T}{1 + \frac{T}{491}},$$

remembering always that T stands for the number of degrees Fahrenheit above the freezing point.

Again, suppose you have a gas at a temperature T above the freezing point, and you want to know what its volume will be at any other temperature T_1: we see that this question may be answered by firstly finding what the volume would be at 32° Fahr., and then, secondly, finding the volume required, but this trouble is unnecessary.

For taking the equation

$$V_{32} \left(1 + \frac{T}{491}\right) = V_T,$$

and a precisely similar one for T_1, viz.

$$V_{32} \left(1 + \frac{T_1}{491}\right) = V_{T_1},$$

and dividing one by the other, we get

$$\frac{1 + \frac{T}{491}}{1 + \frac{T_1}{491}} = \frac{V_T}{V_{T_1}}.$$

The left-hand side of the equation is easily simplified thus:

$$\frac{\frac{491 + T}{491}}{\frac{491 + T_1}{491}} = \frac{V_T}{V_{T_1}}; \quad \therefore \frac{491 + T}{491 + T_1} = \frac{V_T}{V_{T_1}},$$

where T and T_1 are the number of degrees Fahrenheit above the

freezing point; but 491 + the number of degrees above the freezing point amount to just the same thing as 459 + the number of degrees as read on Fahrenheit's thermometer. If, for instance, the temperature of the gas was 50°, it would be 18° above freezing point, and 459 + 50 are just the same as 491 + 18: and our formula may be further simplified by writing the degrees as read on the thermometer + 459 instead of writing it the other way.

Then we get

$$\frac{459 + t_1}{459 + t_2} = \frac{V_1}{V_2}.$$

From this formula we may directly find the volume of the gas at any one temperature when we know the volume at any other.

It is not strictly accurate to say that the expansion is $\frac{1}{491}$ of the volume. Regnault, who made many most careful experiments on the subject, found that for 1° Centigrade for air at atmospheric pressure the expansion was ·003665. This for a degree Fahrenheit is equal to ·002036, or $\frac{1}{491 \cdot 2}$; but the number $\frac{1}{491}$ is most frequently used, and is amply near enough. The expansion of air itself was found by him to be different at different pressures. Thus under a pressure of one atmosphere (or 760 mm.) the coefficient of dilatation through a range of temperature between 32° Fahr. and 212° was found to be ·002039, while it increased gradually as the pressure increased, until, when under a pressure of 79 atmospheres (60,000 mm.), it became ·002507.

Nor are the coefficients of expansion of all gases the same, but for those like hydrogen, oxygen, nitrogen, they are nearly so. When we come to consider carbonic acid the coefficient differs notably; thus comparing carbonic acid and air under 1 atmosphere, that of air is ·002039, and carbonic acid ·002061; while under $3\frac{1}{3}$ atmospheres that of air becomes ·002053, and that of carbonic acid ·002136.

We have seen the effect produced by a simple change of pressure upon a gas, and also the effect produced by a simple change of temperature. We must now see what the effect is when both the temperature and pressure are changed.

Taking the equation given above, $V_1 = V_2 \frac{459 + t_1}{459 + t_2}$, we

will suppose that we know V_2 and t_2 and wish to find V_1. If the pressure at V_1 is double what it was at V_2, the volume would be half. If the pressure at volume V_1 had been $\frac{2}{3}$ of the pressure at the volume V_2, the volume would have been $\frac{3}{2}$, and therefore if we put P_1 and P_2 for the pressures, the volume V_1 becomes

$$V_1 = V_2 \cdot \frac{P_2}{P_1} \cdot \frac{459 + t_1}{459 + t_2},$$

t_1 and t_2 being counted as before from zero Fahr., not from the freezing point.

This formula enables us to tell, from the volume, temperature, and pressure of a gas, what the volume will be at any other temperature and pressure.

Before going on with the subject it may be as well to note what has been termed the absolute zero of temperature.

We have seen that for every degree of temperature which a gas is raised, starting from the freezing point, its volume is increased $\frac{1}{491}$. And it is equally true that for every degree of temperature which a gas is lowered, so far as we can experiment upon them, its volume is diminished approximately $\frac{1}{491}$; so that if gases followed this law perfectly to the extreme limit, it is clear that when the temperature had been lowered 491°, or to − 459° Fahr., its volume would be diminished $\frac{491}{491}$ of its volume at 32° Fahr. This temperature of − 459° has been called the absolute zero of temperature, a temperature at which there is no heat left.

This suggestion, first made I believe by Professor Rankine, is of no value, either theoretical or practical; for, as we have already seen, the coefficients of dilatation are not only slightly different for all gases, but very considerably different in some cases, and, moreover, for the same gas it differs under both different pressures and at different temperatures. So that this absolute zero is not only a shifting point for one and the same gas, but differs with different ones. The gases would also become solid before such a temperature were reached.

We may now go on to examine another matter, viz. the specific heat of gases.

By the specific heat of a gas is meant the amount of heat which must be put into it to alter its temperature a degree as compared

with the amount which must be put into the same weight of water. But in some cases a gas is fixed on as the standard of comparison for all other gases. The specific heat is therefore a ratio, not an absolute quantity. For instance, if it took three times the amount, the specific heat would be 3.

Instead of observing the amount of heat necessary to alter the temperature of the same weight of two gases which are allowed to expand due to the rise of temperature, we may observe the quantity necessary to alter the temperature of the same weight when the volume is not allowed to change. We shall see presently a remarkable relationship existing between the specific heats under these two conditions.

On page 109 in the second column a row of figures is given opposite the names of various gases and vapours, which shows how much heat would be necessary to raise a unit weight, say 1 lb., of each of the gases mentioned, through 1° Fahr. compared with the amount of heat, which is called 1, which would be necessary to raise the same weight of water 1° Fahr.

Running the eye down this column, it will be noticed at once that where a very light gas is taken, such as H, which for equal volumes is only about $\frac{1}{14}$ of the weight of air, the amount of heat required is very large, and that when a very heavy gas like chlorine, which is $2\frac{1}{2}$ times as heavy as air, is taken, then very little heat is required.

From an experimental determination of the specific heats of equal weights of gases and vapours, we may see that the specific heat, multiplied by the specific weight, is in many cases nearly the same quantity. This law is more scientifically stated as the specific heat multiplied by the atomic weight equals a constant quantity, but for our purpose the first expression is perhaps clearer. The specific weight is proportional to the atomic weight.

This law is by no means a strictly accurate one, as the following figures in the first column show: *

It will be noticed that the figures in column 1 are not the product of the figures in column 2, multiplied by those in column 3, but they are proportional to such products. For instance, the atomic weight of nitrogen is 14, and the specific heat from column

* Tyndall, 'Heat as a Mode of Motion,' except for ether and alcohol.

2 is 0·244: these multiplied together give 3·416 instead of 0·237. But if 3·416 be divided by 14·4 we get the number 0·237 as given in column 1; and all the figures in column 1 are found in this way. This number 14·4 is the density of air compared with hydrogen, and has been used in order to keep the figures ·237 opposite air the same in both columns 1 and 2.

	1. Specific Heat of Equal Volumes.	2. Specific Heat of Equal Weights.	3. Atomic Weights.
Water	..	1	
Air	·237	0·237	
Oxygen	·240	0·218	16
Nitrogen	·237	0·244	14
Hydrogen	·236	3·409	1
Chlorine	·296	0·121	
Bromine	·304	0·055	
Nitric acid	·241	0·232	
Carbonic oxide	·237	0·245	
Hydrochloric acid	·235	0·185	
Carbonic acid	·331	0·217	
Nitrous oxide	·345	0·226	
Aqueous vapour	·299	0·480	9
Sulphurous acid	·341	0·154	
Sulphide of hydrogen	·286	0·243	
Bisulphide of carbon	·412	0·157	
Ether vapour	..	0·479	
Alcohol	..	0·453	

We said that the specific heat, or the heat necessary to raise the same weights 1° Fahr. multiplied by the specific gravity, was a constant quantity. Returning to the example of hydrogen and oxygen, if the same weights are taken, the oxygen will only require $\frac{1}{16}$ of the heat required by the hydrogen to raise 1° Fahr., but being sixteen times as heavy, it only occupies $\frac{1}{16}$ of the volume of the hydrogen; so that if the same volume had been taken instead of the same weight, the same amount of heat would have been required in both cases: that is to say, generally that if the statement that the specific heat multiplied by the specific gravity were a true one, then it would be true that equal volumes of gases would require equal amounts of heat to alter their temperature to the same extent.

Chemists have determined the relative weights of the atoms of various gases, and have termed these weights the atomic weights.

They are, as stated before, proportional to the specific gravities o the gases, and the above relationship is usually expressed by saying the atomic weight multiplied by the specific heat is a constant.

Regnault in his work says: "The law which says the specific heats of simple gases referred to the same volume are identical, is an ideal law which might apply to gases following the laws of compression and dilatation, but which is not verified on gases constituted as they are in the circumstances where we are obliged to study them."

Regnault showed also that the specific heat of air varied slightly at different temperatures, thus between $-22°$ Fahr. and $+50°$ Fahr. the specific heat averaged ·23771, between 32° Fahr. and 212° Fahr. it averaged ·23741.

But in the case of carbonic acid Regnault found a very considerable difference. At the freezing point he found it was ·1870, at 212° it became ·2145, and at 392° Fahr. it was ·2396. And the mean specific heat between 50° Fahr. and 410° Fahr. he found to be ·21692.

But when we speak of the amount of heat necessary to alter the temperature 1°, we must be careful to note all the circumstances under which this alteration takes place; for we have seen that any alteration in the temperature of a gas or vapour brings about an alteration in its volume, if the pressure remains the same, and we must see what effect an alteration of volume may have on the amount of heat required to produce any given alteration of temperature. The consideration of this question leads us to the beautiful train of reasoning by which Mayer determined the mechanical equivalent of heat.

Suppose you take a pound weight of air and heat it through 100° Fahr., allowing it to expand and raise the atmosphere. The whole amount of heat is about a quarter of what would have been required to raise 1 lb. of water through 100° Fahr., or more accurately speaking, referring to the Tables of specific heats, it is ·2375 of 100 units, or 23·75 units.

But if, instead of allowing the air to expand as it was heated, we had enclosed it in a vessel of fixed cubical contents, and then heated it, we should have found that instead of requiring 23·75 units of heat to raise it 100° Fahr., we should have required

a considerably smaller quantity. We should in fact have required only 16·73; the heat required when the volume is constant is to the heat required when the pressure is constant as 1 to 1·421.

How does it happen that it takes so much more heat to raise the temperature of the air when it is expanding? The reason is that in expanding it is raising the air, and therefore doing an amount of work which it could not do if it were not heated. When doing work it required 23·75 units, when not doing work 16·73, the difference is 7·02 units. Mayer calculated the work done, and said the amount so done was equivalent to these 7·02 units. Let us follow his calculation: 1 lb. of air occupies a volume of 12·39 cubic feet at the freezing point. Our formula tells us that if it be heated from 32° Fahr. to 132° Fahr., the volume will become $\frac{459 + 132}{459 + 32} \times 12 \cdot 39 = 14 \cdot 92$. The air has therefore increased in volume from 12·39 to 14·92 cubic feet, or by 2·53 cubic feet. We may consider it to have pushed a piston of 1 square foot area through a distance of 2·53 feet, and that the atmospheric pressure was 14·9 lbs. on each square inch. The foot-pounds of work done are consequently the area of the piston, 144 square inches, multiplied by the pressure, or 14·9 lbs. multiplied by the height, or 2·53 feet. This amounts to 5356. The heat required to do this work we found to be 7·02 units, and therefore Mayer said 1 unit was equal to 773·3 foot-pounds.

Dr. Tyndall, in his work on heat, which all engineers should have, gives a most interesting statement of the dates when Mayer and Joule made known their discoveries. He says that Dr. Mayer was a physician in Heilbronn, Germany, and that he first followed this method of calculation in 1842. Mayer in his first paper did not give the details of the calculation, but merely indicated the way he had followed in making it. The actual figure for the equivalent, according to Mayer's calculation, is 771·4.

In 1843 Mr. Joule, from experiments he had made, found the mechanical equivalent to 896, 1001, 1040, 910, 1026, 587, 742 and 860 lbs.; he continued his investigation, and in 1849, after seven years' arduous experimenting, he fixed the equivalence at 772.

Such a near approach to identity as Joule's 772 and Mayer's

771·4, the conclusions having been arrived at in such entirely different manners, is wonderful.

Joule's last experiments in 1849 gave the following results:

772·692	from	friction of	water.	Mean of	40	experiments.
774·083	„	„	mercury	„	50	„
774·987	„	„	cast iron	„	20	„

He, however, considered 772 to be the correct figures.

I will now go on to speak of what is termed the latent heat of steam. When water passes from the liquid to the gaseous state a very large amount of heat has to be put into it after boiling has begun, although the temperature is not altered by this additional quantity. If you begin with a pound of ice-cold water, and gradually heat it up to the boiling point, you will, just when the water begins to boil, have put into it 212 − 32, or 180 units of heat, and the temperature from the moment when the heat was first applied will have continued to rise gradually up to the boiling point: but as soon as boiling begins the temperature ceases to rise, although the water continues to absorb heat, and this heat, which the water continues to absorb until it is all evaporated, is termed the latent heat of steam. It would be found necessary to put in 966·6 units after the water had begun to boil in order to turn it all into steam. So that, starting from the freezing point, and heating the pound of water until it was entirely evaporated at the temperature of water boiling in the air, 180 + 966·6, or 1146·6 units of heat would have been absorbed by the water. This quantity, viz. the amount of heat necessary to turn a unit weight of water at 32° Fahr. into saturated steam, is called the total heat of steam. James Watt, from very limited experimenting, concluded erroneously that the quantity of heat necessary to turn a pound weight of water into saturated steam from the freezing point was the same whatever the pressure. And Southern said that the heat absorbed in the passage from the liquid to the gaseous state was constant for all pressures, and therefore that the total heat was to be found by adding this constant latent heat to the number which represents the temperature of the vapour.

M. Regnault carried out an elaborate series of experiments on the subject, the results of which we shall now give, in which he

proved that the laws of both Watt and Southern were incorrect. His observations extended through a very wide range of temperature, and he found that the formula $\lambda = 606 \cdot 5 + \cdot 305$ T, where λ represents the total heat of steam, and T is in degrees Centigrade, gave results which did not differ from those obtained by experiment more than may be accounted for by errors of observation. Regnault uses the Centigrade scale throughout his work; if, however, Fahrenheit's had been used, then Regnault's formula would become

$$\lambda = 1091 \cdot 7 + 0 \cdot 305\,(T - 32),$$

or simplifying,

$$\lambda = 1081 \cdot 94 + 0 \cdot 305\,T;$$

T being the temperature as read on Fahrenheit's scale (that is, counting from zero and not from 32°), and λ being the total heat, or the heat necessary to convert 1 lb. of water at the freezing temperature into saturated steam at the temperature T.

By means of this formula Table V. was calculated, and by means of the figures at the foot of the table the total heat at any temperature not given may be found, perhaps, more easily than by using the formula directly.

The latent heat is the heat required to turn the boiling water into steam; the total heat includes the heat necessary to raise the temperature of the water from freezing to boiling, supposing you make the freezing temperature your starting point: in the case of water boiling at 212° this amounts to 180 units (212 – 32). If then we deduct the temperature (minus 32) from the total heat, we get the latent heat. For example, the total heat at 400 is 1203·94, therefore the latent heat is 1203·94 – 368 = 835·94, at 212° = 966·6. The Table V. just referred to shows that the latent heat is different for every temperature, and diminishes as the temperature rises, which is at variance with Southern's supposed law.

Regnault further carried out a most minutely accurate series of experiments on the tension of saturated steam at different temperatures. The results of his experiments and the formula by which the tension of the vapour can be calculated for any pressure were given for degrees Centigrade and millimetres.*

* 'Relation des Expériences pour déterminer les principales Lois et les Données numériques qui entrent dans le Calcul des Machines à Vapeur.' Paris, 1847.

The Rev. Robert Dixon, in his work on heat, adapted the formula to the Fahrenheit scale, and inches of mercury, and then calculated the Table III., which, by his kind permission, is copied from that work and given at the end of this book. It shows the tension of the saturated vapour in inches of mercury for every degree Fahrenheit from $-30°$ up to $432°$. See Table III.

(The Rev. R. Dixon has given other valuable tables in his work which I have not availed myself of.)

Regnault's formula, as adapted by Dixon, is

$$\log. i = a - b\,\alpha^{T} - c\,\beta^{T},$$

where i = inches of mercury at the equator at the sea level, T = degrees Fahrenheit counted from 32 upwards.

And for convenience in calculating he gives the logarithms of the above constants:

$$a = 4{\cdot}885\ 984\ 524\ 7$$
$$\log. \alpha = \bar{1}{\cdot}999\ 079\ 751\ 3$$
$$\log. \beta = \bar{1}{\cdot}996\ 693\ 778\ 3$$
$$\log. b = 0{\cdot}659\ 317\ 975\ 2$$
$$\log. c = 0{\cdot}020\ 517\ 432\ 4$$

This formula was intended by Regnault to hold good between the limits of temperature $-30°$ Fahr. and $450°$ Fahr.*

There is another subject of especial interest to engineers connected with steam, and this is the relationship of the volume occupied by the steam to the volume occupied by the water from which it was generated, or to, what is the same thing, the volume of water to which the steam would condense. This ratio is termed the specific volume.

The most valuable investigation into this subject is that conducted by Messrs. Fairbairn and Tate. They published the results in the 'Philosophical Transactions' for 1860 and 1862, from which I have taken in full the figures given in the three first columns of the following table, except the pressure corresponding to 212° Fahr. The meaning of the other columns will be explained later on.

* In Regnault's work other formulæ are given, but the above is the most useful, and for all practical purposes the table at the end of this book is sufficient.

1. Pressure in Inches of Mercury.	2. Maximum Temperature of Saturation in Degrees Fahr.	3. Specific Volume of Steam determined by Experiment.	4. Work done by Steam in raising the Pressure through the Specific Volume, in Units of Heat.	5. Total Quantity of Heat as determined by Regnault's Formula: $\lambda = 1081 \cdot 94 + \cdot 305\ T$.	6. Apparent True Total Heat found by deducting figures in Column 4 from those in Column 5.	7. Specific Volume of Steam calculated from the formula $Sv = \frac{\cdot 0649T + 57 \cdot 426}{\cdot 001465 \cdot P}$.
5·35	136·77	8275·3	64·839	1123·65		8641
8·62	155·33	5333·5	67·176	1129·3	1062·12 A	5347
9·45	159·36	4920·2	68·094	1130·6	1062·5 B	4896
12·47	170·92	3722·6	67·984	1134·07	1066·09 C	3752
12·61	171·48	3715·1	68·608	1134·24	1065·63 D	3712
13·62	174·92	3438·1	68·579	1135·29	1066·71 E	3448
16·01	182·30	3051·0	71·536	1137·54	1066·0 F	2954
18·36	188·30	2624·4	70·54	1139·37	1068·83 G	2590
22·88	198·78	2149·5	72·026	1142·57	..	2098
29·898	212	...	..	1146·6	..	1626
53·61	242·90	943·1	74·045	1156·02	..	932
55·52	244·82	908·0	73·829	1156·61	..	901·6
55·89	245·22	892·5	73·053	1156·73	..	895·9
66·84	255·50	759·4	74·337	1159·87	..	756
76·20	263·14	694·2	72·448	1162·19	..	667·5
81·53	267·21	653·3	78·005	1163·43	..	626·1
84·20	269·20	605·7	74·69	1164·04	1089·35 H	607·3
90·08	273·30	543·2	71·661	1165·3	1093·64 I	569·7
92·23	274·76	584·4	78·936	1165·74	1086·8 J	557·2
99·60	279·42	515·0	75·121	1167·16	1092·04 K	517·9
04·54	282·58	497·2	76·157	1168·13	1091·97 L	495
12·78	287·25	458·3	75·697	1169·54	1093·84 M	460·5
14·25	288·25	449·6	75·228	1169·84	1094·61 N	454·8
22·25	292·53	433·1	77·541	1171·16	1093·62 O	426·5
Taken in full from the 'Phil. Trans.,' 1860.						

On the results of their experiments, which were conducted in a most ingenious manner, Messrs. Fairbairn and Tate based the following formula:

$$V = 25 \cdot 62 + \frac{49513}{P + \cdot 72},$$

and

$$P = \frac{49513}{V - 25 \cdot 62} - \cdot 72,$$

where V is the specific volume and P the tension of the steam measured by a column of mercury in inches.

If we substitute pounds for inches of mercury the formula becomes

$$V = 25 \cdot 62 + \frac{24260}{P + \cdot 353}.$$

* The figures in this row were not given in the 'Phil. Trans.'

From this formula we may calculate the specific volume for any tension in pounds.

Other formulæ have been given for V, but the above one being based upon the most carefully conducted experiments, and agreeing very nearly with accurate experimental results, appears to be the most trustworthy.

The figures given in that investigation are so exceedingly interesting that I propose to examine them in the following pages at some length, for from those results we may deduce another formula for the specific volume of steam from which Table IV. has been calculated, as well as some facts connected with the specific heat of steam.

When water is being boiled into steam at any tension, heat has to be added for two perfectly distinct reasons, the one of which is the change of state from liquid to vapour, the other is the raising of the pressure exerted by the vapour itself. If you boil off water at a tension of 60 lbs., which roughly corresponds to a boiler pressure of 45 lbs., every time a cubic inch of vapour is formed it has while being formed to force back a pressure of 60 lbs. per square inch through a distance of 1 inch. As you cannot do work for nothing, you must put an extra amount of heat (beyond that merely required to turn a liquid into a vapour) into the steam. What this amount is may be calculated in the following manner. One pound of water occupies 27·69 cubic inches; suppose it to be contained in a tube of 1 square inch sectional area, and to be converted into steam at a pressure of P lbs. on the square inch, and to occupy a specific volume V,

Then the foot-pounds of work done in raising the pressure through the specific volume are $P \times (V - 1) \frac{27 \cdot 69}{12}$. We divide by 12 of course, because our volumes are in inches, and we wish them in feet, and we say $V - 1$ because the pressure is not lifted through the whole specific volume by the amount previously occupied by the water; but inasmuch as V is for all moderate ranges of temperature very large compared with 1, we may with sufficient practical accuracy say that the work done in foot-pounds is

$$p \times V \times \frac{27 \cdot 69}{12},$$

and therefore the units of heat necessitated are

$$p \times V \times \frac{27 \cdot 69}{12 \times 772} = p \times V \times \cdot 002989,$$

where V is the specific volume and p the pressure in pounds.

In Messrs. Fairbairn and Tate's experiments the pressures were given in inches; therefore adapting our formula to mercury inches, we get $p \times V \times \cdot 001465$, V being the specific volume and p the pressure in inches of mercury.

By means of this formula the figures in column 4 were calculated, page 115. They are the number of heat units equivalent to the work done by the steam from 1 lb. of water in raising the pressure through the volume occupied by the steam.

Now the figures in column 5 are calculated according to Regnault's formula for the total heat, already given on page 113, viz. $\lambda = 1081 \cdot 94 + \cdot 305\,T$.

The figures in column 5 show the total heat necessary to turn a pound of water into steam; the figures in column 4 show how much of that heat was utilized in forcing back the pressure. If we deduct this latter from the former, we get what I have called the true total heat of the steam at that temperature, that is to say, the heat necessary to turn a pound of water from the freezing point into steam without allowing the steam to do any external work, or, what is precisely the same thing, putting the pound of water into a vacuous space (equal to the specific volume multiplied by $27 \cdot 69$), and then turning it into steam.

The figures in column 6 have been so found, and have, for the sake of reference, been figured A, B, C, D, &c. These quantities I have termed the apparent true total heats.

The specific heat of steam may then be found as follows: From column 2 we may take two temperatures, and subtracting the less from the greater, we get a number representing a certain rise of temperature. From column 6 we get, opposite the two chosen temperatures, figures representing the absolute amount of heat in the steam at those two temperatures; deducting the less from the greater, we get the amount of heat which has been added corresponding to the rise of temperature previously found from column 2. Now, by dividing the total amount of heat added by the

number of degrees rise of temperature occasioned by this addition, we get a specific heat of steam. Thus, taking the cases marked A and H in column 6, we see that the total heat added is, $1089 \cdot 35 - 1062 \cdot 12 = 27 \cdot 23$, and column 2 tells us the rise of temperature is $269 \cdot 20 - 155 \cdot 33 = 113 \cdot 87$; dividing the former by the latter, we get ·23913.

We may then try A and I, A and J, &c., then B and H, B and I, &c., and so on for them all. The following table shows each of the quantities so found, and from what data:

	A	B	C	D	E	F	G
H	·23913	·24445	·23667	·24273	·24014	·26870	·25356
I	·26719	·27330	·26971	·27509	·27373	·30374	·29188
J	·20665	·21057	·19944	·20498	·20122	·22496	·20784
K	·24111	·24604	·23917	·24524	·24239	·26813	·25472
L	·23458	·23917	·23177	·23709	·23517	·25898	·24544
M	·24045	·24505	·23855	·24367	·24152	·26527	·25275
N	·24444	·24971	·24308	·24818	·24618	·27003	·25793
O	·22959	·23369	·22638	·24785	·22881	·25057	·23784

The mean of all the fifty-six quantities is ·24457; but if we throw out all the figures in the columns opposite the letters I J F G as being apparently extreme values, the mean of the remaining thirty quantities becomes ·2400667, or ·2401.

·2401 may therefore be taken as an approximate value of what I have, in this examination of Messrs. Fairbairn and Tate's experimental results, called the specific heat of steam.

At this point I make an assumption which is a true one for a perfect gas, but I am not aware whether steam has been experimented on or not. I assume that saturated steam, when expanding without doing any work, that is, expanding into a vacuum, undergoes no change of temperature.

Gay-Lussac first proved this to be the case for air. He had two vessels in communication by means of a short pipe with a cock on it between them. In one of these vessels there was air, in the other nothing, or, in engineering parlance, a vacuum. He opened the cock, and allowed the air to pass from one vessel to the other until the pressure was equal in each, and found no change in tem-

perature. Joule went farther: he compressed air to a pressure of 20 atmospheres, and surrounded both vessels with water before opening the cock. After the cock had been opened, and the pressure was the same in both vessels, no change of temperature was observable.

Suppose we consider this to be true for steam. Regnault's formula says that the total heat of steam at any temperature $T_1° = 1081 \cdot 94 + \cdot 305\ T_1$. Now calling $S\ v_1$ and p_1 the specific volume, and pressure in inches of mercury corresponding to $T_1°$, then what we have termed the true total heat at any temperature

T_1 becomes $= 1081 \cdot 94 + \cdot 305\ T_1 - S\,v_1 \times p_1 \times \cdot 001465$. (See page 117.) [A]

and at

$$T_2 \text{ it becomes} = 1081 \cdot 94 + \cdot 305\ T_2 - S\,v_2 \times p_2 \times \cdot 001465. \qquad [B]$$

A pound of saturated steam formed in a vacuum at a temperature T_2 and a pressure of P_2 mercury inches would contain an amount of heat which formula B gives us. Now let this steam increase its volume in a vacuum (that is, without doing work, as in Joule's experiment) until it occupies the volume of 1 lb. of saturated steam at a temperature $T_1°$. If our assumption is a correct one there will be no alteration of temperature, which will remain $T_2°$, and of course there will be no alteration in the total amount of heat contained. Consequently the pound of steam at its new volume is superheated, for to retain it as a saturated vapour the temperature should be $T_1°$, whereas it is $T_2°$. It is superheated $T_2° - T_1°$, and therefore contains $(T_2° - T_1°) \times \cdot 2401$ units of heat more than is necessary to hold it as a saturated vapour. If therefore we deduct $(T_2° - T_1°) \cdot 2401$ units of heat from the amount of heat contained in the saturated vapour at $T_2°$, we shall get the amount of heat contained in the saturated vapour at $T_1°$. Now the formulæ A and B give us these two quantities. Therefore we say

$$\begin{aligned} 1081 \cdot 94 + \cdot 305\ T_2 - \cdot 001465\ S\,v_2 \times p_2 - \cdot 2401\,(T_2 - T_1) \\ = 1081 \cdot 94 + \cdot 305\ T_1 - \cdot 001465\ S\,v_1 \times p_1, \end{aligned}$$

and therefore

$$\begin{aligned} \cdot 305\,(T_2 - T_1) - \cdot 2401\,(T_2 - T_1) - \cdot 001465\ S\,v_2 \times p_2 \\ = - \cdot 001465\ S\,v_1 \times p_1; \end{aligned}$$

and therefore

$$\cdot 0649\,(T_2 - T_1) + \cdot 001465\, S v_1 \times p_1 = \cdot 001465\, S v_2 \times p_2,$$

$$\therefore \frac{\cdot 0649\,(T_2 - T_1) + \cdot 001465\, S v_1 \times p_1}{\cdot 001465\, p_2} = S v_2.$$

If we can assign any value to $S\,v_1 \times p_1$, we have a formula which gives us the specific volume corresponding to any pressure.

To do this with the nearest approximation to correctness, the specific volume at a temperature of 300° Fahr. and corresponding pressure of 136·72 inches of mercury was calculated for 22 out of the 23 experimentally found data given by Messrs. Fairbairn and Tate, T_1 $S\,v_1$ and p_1 being each time taken from columns 1, 2, and 3. The result was a mean specific volume of 385; throwing out 5 extreme values, it becomes 384.

Now, let us substitute this value in the above equation for $S\,v_1$, 136·72 for p_1, and 300 for T_1, and our formula becomes

$$S v_2 = \frac{\cdot 0649\,(T_2 - 300) + \cdot 001465 \times 136\cdot 72 \times 384}{\cdot 001465\, p_2},$$

or

$$S v_2 = \frac{\cdot 0649\, T_2 + 57\cdot 42}{\cdot 001465\, p_2},$$

and dropping accents

$$S v = \frac{\cdot 0649\, T + 57\cdot 42}{\cdot 001465\, p},$$

p being in inches of mercury, T the number of degrees Fahr., counting from zero, not from the freezing point.

By means of this formula the specific volume for all the temperatures at which Messrs. Fairbairn and Tate experimented, were calculated, and the results given in column 7. These quantities so calculated will be found to be in moderately close accord with those found experimentally and given in column 3.

Our general formula

$$S v = \frac{\cdot 0649\, T + 57\cdot 42}{\cdot 001465 . p}$$

is the most convenient form, because p can be found corresponding to any temperature T, from Professor Dixon's Table, given at the

end of this work. p is there in inches of mercury, but if pounds per square inch were the unit, then the formula would become

$$S\,v = \frac{\cdot 0649\,T + 57\cdot 42}{\cdot 00299 \times p}$$

A Table of calculated specific volumes will be found at the end of the work. See Table IV.

We saw on page 119, that if saturated steam at a temperature of T_2° and volume $S\,v_2$ were allowed to expand to $S\,v_1$, without doing work, it would superheat itself $T_2^\circ - T_1^\circ$ of sensible heat, and would therefore superheat itself to the extent of $(T_2 - T_1)\cdot 2401$ units of heat. If we call H_2 and H_1 the true total heats, see p. 119, then

$$H_2 - (T_2 - T_1)\cdot 2401 = H_1,$$

therefore

$$H_2 - T_2\cdot 2401 = H_1 - T_1\cdot 2401;$$

and therefore generally

$$H_n - T_n\cdot 2401 = H_1 - T_1\cdot 2401,$$

where T_n is any temperature whatever; and therefore

$$H_n - T_n\cdot 2401 = C, \text{ or a constant quantity.}$$

We may write this

$$H_n = C + T_n\cdot 2401.$$

This means that the heat required to turn water at 32° Fahr. in a vacuum, into steam, is for all temperatures and corresponding pressures a constant quantity plus $\cdot 2401$ the degrees of sensible temperature. We may easily see what this constant quantity C is; our formula says

$$S\,v = \frac{\cdot 0649\,T + 57\cdot 42}{\cdot 001465 \times p},$$

therefore

$$S\,v \times \cdot 001465 \times p = \cdot 0649\,T + 57\cdot 42;$$

but the left-hand member of this equation is the work done in forcing back the vapour, therefore the right-hand member is also equal to that; and since $\lambda = H_n$ + work done in raising the vapour, therefore

$$\lambda = H_n + \cdot 0649\,T_n + 57\cdot 42,$$

but Regnault's formula says

$$\lambda = 1081\cdot 94 + 0\cdot 305\,T_n$$

$$\therefore\ 1081\cdot 94 + 0\cdot 305\,T_n = H_n + \cdot 0649\,T_n + 57\cdot 42,$$

and again

$$H_s = C + T_s \times \cdot 2401$$

$$\therefore\ 1081 \cdot 94 + 0 \cdot 305\, T_s = C + T_s \times \cdot 2401 + \cdot 0649\, T_s + 57 \cdot 42,$$

wherefore

$$C = 1024 \cdot 51.$$

The knowledge we have obtained of the relation of the temperature to the pressure and specific volume of steam will enable us to see at once that if steam expanded doing the full amount of work corresponding to the varying pressure, as steam does when expanding in a cylinder, partial condensation must inevitably ensue in a cylinder made of a perfect non-conducting material, that is, a cylinder which does not give up any heat to the steam. One example, based entirely upon experiment, will suffice to prove this. The dotted black curved line in Fig. 52 is the curve of specific volumes laid down from Fairbairn and Tate's experiments. The vertical ordinates represent the pressure in pounds, the horizontal ordinates the volumes occupied by the steam at the different pressures.

We will take 1 lb. of steam at a specific volume 2149·5, and suppose it, as we have frequently done before, to push back a piston of 1 square inch area, until it occupies a specific volume 5335. The mean pressure between these specific volumes A and B may be found from the figure to be close upon 6·5 lbs. Therefore, 6·5 lbs. have been raised through $(5333 - 2149) \times 27 \cdot 7 = 3184 \times 27 \cdot 7$ inches, and the number of units of heat necessary to do this are $\frac{3184}{12 \times 772} \times 6 \cdot 5 \times 27 \cdot 7 = 61 \cdot 9$.

If the steam in expanding from A to B never condensed, then the steam must always have been either in the saturated state, and must have shown pressures as given by the line A B, or the steam must have been superheated, in which case the pressures would have been above the line A B. If the pressures had been above the line A B, then in expanding the steam would have done more work than is equivalent to 61·9 units. We will consider it as doing 61·9 units only, which is the lowest possible quantity compatible with the assumption that no condensation takes place.

Now when steam occupies a specific volume of 2149·5 it con-

tains 1142·57 units of heat; now it expands and loses 61·9 units, and therefore when it reaches a specific volume of 5333·5 at B it contains 1080·67 units.

But to maintain steam in a saturated condition at a specific volume of 5333·5 you must have 1129·3 units, and you have only got 1080·67. The steam cannot therefore have done as much work as the line A B represents, and must therefore have partially condensed.

If the curved line A B had been drawn in according to Boyle's law, starting from the point A, it would have been slightly above its present position, and would therefore have represented a still larger expenditure of heat units while expanding, and a consequently greater condensation, than in the first case considered.

It is still occasionally maintained that steam in expanding does not condense; on this account the condensation has not been assumed.

We see at once that if saturated steam in expanding partially condenses, it must be superheated if compressed; for if it be not superheated by the compression let it remain saturated, then allow it to expand again, and we know condensation will result; whereas we have by expansion only taken out at most as much heat as we put in during the compression, and there was enough heat originally in it to hold it as a saturated vapour, and there should therefore be that amount still in it.

That a very large amount of condensation does take place in both compound and simple expansive engines is proved by the most unquestionable testimony. At the end of this chapter is given an account of the trials of three American vessels, the 'Rush,' the 'Dexter,' and the 'Dallas.'

The amount of steam which disappeared partly by condensation, and to a certain extent by leakage, will be seen to have been, in the cases of the non-compound engines, enormous; but I do not wish it to be inferred that all the steam which is condensed, is condensed owing to the abstraction of heat while the steam is working in the cylinder. Other causes of condensation have been referred to.

Although a pound weight of steam cannot expand following Boyle's law, nor even expand showing pressures considerably below

those given by Boyle's law, yet we do find that, in a very large number of engines, the expansion, as shown by the indicator card, has taken place in very close accordance with that law. The fact is, that a considerable quantity of the steam when first admitted to the engine at the beginning of the stroke is at once condensed, and when the cut off has taken place and the pressure begins to fall this condensed water, which is at a high temperature, begins to boil, so that the quantity of steam in the cylinder is increased as the piston nears the exhausting point. If it had not been for this re-boiling the pressure would have been much below the point where we find it.

A gas may expand under two very different influences, and the indicator cards they would give are known by the two names Isothermic and Adiabatic. When the gas expands and its temperature is kept constant throughout the experiment, the curve is an isothermic one; when it expands, no heat being either put into it or taken out of it, the curve becomes an adiabatic one. In an adiabatic curve the heat lost by the gas while doing work is not returned to it; in an isothermic curve it is. In Professor Clerk-Maxwell's book on heat, the interesting subject of these curves has been treated at some length, and the reader is referred to it for further information.

A very large amount of ingenuity has been expended in the attempt to discover the law which steam follows in expanding. The fact remains, however, that for practical purposes not one of them can compare in point of utility with the commonly used one of Boyle and Mariotte. Even supposing any such formula to be found, which would accurately give the shape of the expansion portion of the indicator card for each separate engine, it still only determines one small portion of the card, and that by no means the most important. It would not tell us how much to deduct from the boiler pressure to find the pressure in the cylinder; it would not tell us the shape of the card where the wire-drawing cut off took place, nor the shape of the exhaust corner, nor, more important than all, would it tell us what vacuum you were going to get in the cylinder; so that as a guide to the probable indicated horse-power of an engine Boyle and Mariotte's law may be considered as infinitely the simplest, and, considering on how many

other things the power is dependent, it may also be considered as quite sufficiently accurate.

I will just notice one of the simplest of these formulæ, viz.

$$P V^{\frac{10}{9}} = C,$$

Boyle's law saying

$$p \times V = C.$$

Taking this first, since

$$p = \frac{C}{V},$$

the area of the curve, viz.

$$\int p\, d v = C \log._{\epsilon} V,$$

generally, now let the steam expand to any new volume V^1, the area of the curve is again

$$= C \log._{\epsilon} V^1,$$

$\therefore$ the area of the curve between the two points V and V, that is during the expansion, is

$$C \log._{\epsilon} V^1 - C \log._{\epsilon} V = C (\log._{\epsilon} V^1 - \log._{\epsilon} V)$$
$$= C \log._{\epsilon} \frac{V^1}{V},$$

or, using the common logarithms, we get

$$= C \times 2 \cdot 3026 \times \log. \frac{V^1}{V}. \qquad [A]$$

Now taking the formula

$$p \times V^{\frac{10}{9}} = C,$$

therefore

$$p = C \frac{1}{V^{\frac{10}{9}}} = C V^{-\frac{10}{9}} \text{ and } \int p\, d v = C \int V^{-\frac{10}{9}} d v.$$

Now generally

$$\int x^m d x = \frac{x^{m+1}}{m+1},$$

except where $m = -1$.

Write $-\frac{10}{9}$ for m, and

$$\int p\,d\,v = \mathrm{C}\,\frac{\mathrm{V}^{-\frac{1}{9}}}{-\frac{1}{9}} = -9\,\mathrm{C}\,\frac{1}{\mathrm{V}^{\frac{1}{9}}},$$

and for any new volume

$$\mathrm{V}_1 = -9\,\mathrm{C}\,\frac{1}{\mathrm{V}_1^{\frac{1}{9}}};$$

$$\therefore \text{ the area during expansion is } -9\,\mathrm{C}\left(\frac{1}{\mathrm{V}_1^{\frac{1}{9}}} - \frac{1}{\mathrm{V}^{\frac{1}{9}}}\right). \qquad [\mathrm{B}]$$

We must remember that the two C's which appear in A and B are not the same. We will now take a cylinder cutting off at 12 inches and exhausting at 96, which is an expansion of eight times, with an initial tension of 60 lbs. per square inch.

Equation A says that during expansion the area of the curve is

$$\mathrm{C} \times 2\cdot3026 \times \log.\ 8 = 60 \times 12 \times 2\cdot3026 \times \cdot90309;$$

and if we divide this by $96 - 12 = 84$ we get the mean pressure in lbs. $= 17\cdot8$.

In formula B,

$$\mathrm{C} = p\,.\,\mathrm{V}^{\frac{10}{9}} = 60 \times 15\cdot8,$$

and the area

$$= -9 \times 60 \times 15\cdot8\left(\frac{1}{\mathrm{V}_1^{\frac{1}{9}}} - \frac{1}{\mathrm{V}^{\frac{1}{9}}}\right)$$

$$= -9 \times 60 \times 15\cdot8\,(\cdot6022 - \cdot7587)$$

$$= -9 \times 60 \times 15\cdot8\,(-\ \cdot1565),$$

and dividing by 84 the length of the curve, our mean pressure becomes $15\cdot8$.

When it is desirable to find out the probable mean pressure, the author strongly recommends the draughtsman to make a drawing of the card, and not on any account to trust to any formula which has been yet given.

TRIALS OF THE 'RUSH,' 'DEXTER,' AND 'DALLAS.'

In the year 1874 a trial of the greatest interest to all engineers was conducted in America upon three Government vessels, the 'Rush,' the 'Dexter,' and the 'Dallas.' The trial was conducted by Mr. C. H. Loring, Chief Engineer of the United States Navy, and Mr. C. E. Emery, consulting engineer, with great care, giving us comparable results of a perfectly reliable nature. Not only are the results they obtained of great value, but the manner in which they set about obtaining them, the exceedingly careful manner in which the whole investigation was made, constitutes a valuable lesson to engineers generally.

The engine of the 'Rush' was a compound one, with vertical cylinders with intermediate receiver, taking on to cranks at right angles. The cylinders were jacketed, 24 and 38 inches diameter and 27 inches stroke. Surface condenser.

The 'Dexter's' engine was a single vertical cylinder, 26 inches diameter and 36 inches stroke. It was not jacketed. Surface condenser.

The 'Dallas's' engine was a single vertical cylinder, 36 inches diameter, with 30 inches stroke. The cylinder was not steam jacketed. Surface condenser.

Manner of making the Experiments. (Copied from Messrs. Loring and Emery's Report.)

The experiments were made with vessels secured to the wharf. The coal, which was anthracite of fair quality, was broken on the wharf to proper size (the vessels' bunkers having been closed and sealed), and filled into bags to a certain weight. The bags were sent on board when ordered by the senior engineer on watch, he making record on the log of the number of bags and the time of receipt, a similar record being made by one of the men on the wharf. At the end of the hour the number of bags of coal actually put on the fire, was reported from the fire-room, and entered in the appropriate column. The several records agreed

with each other, and the total amount expended corresponded with the total number of bags filled on the wharf. The ashes were measured into buckets (of which the mean weight was ascertained), and tallied as they were hoisted out. They were afterwards weighed in the gross on the wharf, and the two accounts found to agree substantially.

The feed water was measured after its delivery from the surface condenser and before its return to the boiler, for which purpose a tank of boiler plate was especially constructed, having a plate dividing it vertically into two equal parts. In the upper edge of the plate was cut a rectangular notch 8 inches long, by which the height to which each half of the tank could be filled was determined. The mean of the weight of water which the half-tank contained was 1129½ lbs., at a temperature of 72° Fahr.

In the computations for each experiment, the weight of water is reduced to correspond with mean temperature.

One of the feed pumps was disconnected from the check-feed valve, and its discharge pipe led to a small receiving tank, placed over the two halves of the measuring tank, into which this pump forced the condensed water from the hot well. The receiving tank had on its bottom two cocks, one over each half-tank, so that either could be filled from it at will. The other feed pump had its suction pipe detached from the hot well, and connected with the bottoms of the two half-tanks through a cock on each, so that the contents of either could be drawn out and discharged into the boiler.

The method of measuring the water and recording it was as follows: one side having been filled, the cock over it on the receiving tank was closed, and the other over the empty half opened. When the water in the full one had settled to the height of the edge of the notch, its cock in the feed pipe was opened, and the contents pumped into the boiler (care being taken to empty one in less time than it took to fill the other). When empty, its feed cock was closed. When the water in the tank being filled reached within a few inches of the notch, a gong in the engine-room was sounded to call attention, and when it reached the notch the gong was struck twice. At this instant the assistant-engineer in the engine-

room noted the reading of the counter, and an attendant in the fire-room noted and reported the height of water in the glass guage in the boiler, as shown by a scale of inches secured to it. The attendant at the tank also noted the time of filling, and the temperature when the tank was half emptied. After entering the number of the counter in the log, the assistant-engineer ascertained the numerical difference between that and the preceding entry, and if it was far from the average its cause was sought for.

By this system of checks all errors of record could be detected, and it was possible to preserve and utilize any continuous run which came to an end through derangement of the engine. All parts of the tanks, pipes, and cocks were plainly visible to the eye; and had any leaks occurred therein, they must have been detected. That the condensers were tight was evident from the fact that the water remained quite fresh in the boilers.

The water lost from ordinary causes in the circulation to and from the engine and boiler, was replaced by running hydrant water in the tank that was being filled. The additional water was therefore measured and charged in the cost.

The loss of water was not sufficient to affect the result materially in either case. It was greatest in the 'Dexter,' which had been on service. The safety valve of this vessel leaked slightly, and there was probably some other trifling leak that could not be detected. The number of inches that the water fell in the boiler between periods of supply being shown in the logs, were added together, and from the same and the known dimensions of the boiler, the volume and weight lost were ascertained quite accurately. The reduction in the number of revolutions per tank, when the water was being received from the hydrant, furnished another and perhaps still more accurate means of ascertaining the proportionate amount lost and returned. The two methods closely agreed in fixing the loss in the case of the 'Dexter' at 4·96 per cent. of the total amount of water used.

A number of indicators were tested with steam before the trials, and a pair selected for use which proved correct by a standard gauge at varying pressures. Indicator diagrams were taken every

twenty minutes throughout the trials, and the data of the usual columns of the log (except the coal and ashes) every half hour.

It was ascertained that the pistons of the 'Dexter' and 'Dallas' were tight by removing the cylinder covers, and letting on full steam pressure.

During the first and principal experiments with each vessel, the several boilers were worked at their maximum power with natural draught at the dock, the fires being cleaned regularly as at sea, and the cut-offs adjusted to carry a steam pressure of about 70 lbs. during the trial of the 'Rush' and 'Dexter,' and about 35 lbs. during the trial of the 'Dallas.'

Table showing the performances of the three U.S. vessels, 'Rush,' 'Dexter,' and 'Dallas,' condensed from the Table given in Messrs. Loring and Emery's Report.

	'Rush.'		'Dexter.'	'Dallas.'
	h.p. cyl.	l.p. cyl.		
Duration of trial	55 hours.		34½ hrs.	31 hrs.
Pounds of water used to 1 I.H.P. per hour as measured from tank	18·38		23·9	26·9
Percentage of water used accounted for by the indicator	93·23	73·55	68·32	74·45
Percentage of water which does not make its appearance in the indicator	6·8	26·45	31·68	25·5*
Number of lbs. of water evaporated at the observed temperature and pressure ..	7·55		7·63	7·86
Coals consumed per I.H.P. per hour ..	2·43		3·13	3·42
Coals burnt per square foot per hour ..	11·4		12	13·3
Mean I.H.P.	266·5		219	221·5
Ratio of expansion	6·22		3·49	3·13
Piston speed	319		366	308*
Steam	69		67	32
Vacuum	26·5		25·5	25·2
Diam. cylinders in inches	24	38	26	36
Stroke in inches	27		36	30
Total grate surface in feet	57		57	57
„ heating surface in feet	1573		1573	1689
„ condensing surface in feet	900		900	900

* This row of figures is added by the author.

TABLE I.

Barometric Scale in Millimetres and Inches.†

Mm.	Inches.	Mm.	Inches.	Mm.	Inches.	Mm.	Inches.
700	= 27·560	723	= 28·465	746	= 29·371	768	= 30·237
701	= 27·590	724	= 28·504	747	= 29·410	769	= 30·276
702	= 27·638	725	= 28·543	748	= 29·449	770	= 30·315
703	= 27·678	726	= 28·583	749	= 29·489	771	= 30·355
704	= 27·717	727	= 28·622	750	= 29·528	772	= 30·384
705	= 27·756	728	= 28·661	751	= 29·567	773	= 30·434
706	= 27·795	729	= 28·701	752	= 29·607	774	= 30·473
707	= 27·835	730	= 28·741	753	= 29·646	775	= 30·512
708	= 27·876	731	= 28·780	754	= 29·685	776	= 30·552
709	= 27·914	732	= 28·819	755	= 29·725	777	= 30·591
710	= 27·953	733	= 28·859	756	= 29·764	778	= 30·631
711	= 27·992	734	= 28·898	757	= 29·804	779	= 30·670
712	= 28·032	735	= 28·938	758	= 29·843	780	= 30·709
713	= 28·071	736	= 28·977	759	= 29·882	781	= 30·749
714	= 28·111	737	= 29·016	760	= 29·922	782	= 30·788
715	= 28·150	738	= 29·056	761	= 29·961	783	= 30·827
716	= 28·189	739	= 29·095	762	= 30·000	784	= 30·867
717	= 28·229	740	= 29·134	763	= 30·040	785	= 30·906
718	= 28·268	741	= 29·174	764	= 30·079	786	= 30·945
719	= 28·308	742	= 29·213	765	= 30·119	787	= 30·985
720	= 28·347	743	= 29·252	766	= 30·158	788	= 31·024
721	= 28·386	744	= 29·292	767	= 30·197	789	= 31·063
722	= 28·426	745	= 29·331				

† Copied from Brande and Taylor's 'Chemistry.'

TABLE II.

Thermometrical Equivalents.*

Cent.	Fahr.	Cent.	Fahr.	Cent.	Fahr.	Cent.	Fahr.	Cent.	Fahr.
−30	−22·0	3	37·4	36	96·8	69	156·2	102	215·6
−29	−20·2	4	39·2	37	98·6	70	158·0	103	217·4
−28	−18·4	5	41·0	38	100·4	71	159·8	104	219·2
−27	−16·6	6	42·8	39	102·2	72	161·6	105	221·0
−26	−14·8	7	44·6	40	104·0	73	163·4	106	222·8
−25	−13·0	8	46·4	41	105·8	74	165·2	107	224·6
−24	−11·2	9	48·2	42	107·6	75	167·0	108	226·4
−23	− 9·4	10	50·0	43	109·4	76	168·8	109	228·2
−22	− 7·6	11	51·8	44	111·2	77	170·6	110	230·0
−21	− 5·8	12	53·6	45	113·0	78	172·4	111	231·8
−20	− 4·0	13	55·4	46	114·8	79	174·2	112	233·6
−19	− 2·2	14	57·2	47	116·6	80	176·0	113	235·4
−18	− 0·4	15	59·0	48	118·4	81	177·8	114	237·2
−17	+ 1·4	16	60·8	49	120·2	82	179·6	115	239·0
−16	+ 3·2	17	62·6	50	122·0	83	181·4	116	240·8
−15	+ 5	18	64·4	51	123·8	84	183·2	117	242·6
−14	+ 6·8	19	66·2	52	125·6	85	185·0	118	244·4
−13	+ 8·6	20	68·0	53	127·4	86	186·8	119	246·2
−12	+10·4	21	69·8	54	129·2	87	188·6	120	248·0
−11	+12·2	22	71·6	55	131·0	88	190·4	121	249·8
−10	+14·0	23	73·4	56	132·8	89	192·2	122	251·6
− 9	+15·8	24	75·2	57	134·6	90	194·0	123	253·4
− 8	+17·6	25	77·	58	136·4	91	195·8	124	255·2
− 7	+19·4	26	78·8	59	138·2	92	197·6	125	257·0
− 6	+21·2	27	80·6	60	140·0	93	199·4	126	258·8
− 5	+23·0	28	82·4	61	141·8	94	201·2	127	260·6
− 4	+24·8	29	84·2	62	143·6	95	203·0	128	262·4
− 3	+26·6	30	86·0	63	145·4	96	204·8	129	264·2
− 2	+28·4	31	87·8	64	147·2	97	206·6	130	266·0
− 1	+30·2	32	89·6	65	149·0	98	208·4	131	267·8
0	+32·0	33	91·4	66	150·8	99	210·2	132	269·6
1	+33·8	34	93·2	67	152·6	100	212·0	133	271·4
2	35·6	35	95·	68	154·4	101	213·8	134	273·2

* Copied from Brande and Taylor's 'Chemistry.'

TABLE II.—*continued.*

Cent.	Fahr.	Cent.	Fahr.	Cent.	Fahr.	Cent.	Fahr.	Cent.	Fahr.
135	275·0	169	336·2	202	395·6	235	455·0	268	514·4
136	276·8	170	338·0	203	397·4	236	456·8	269	516·2
137	278·6	171	339·8	204	399·2	237	458·6	270	518·0
138	280·4	172	341·6	205	401·	238	460·4	271	519·8
139	282·2	173	343·4	206	402·8	239	462·2	272	521·6
140	284·0	174	345·2	207	404·6	240	464·0	273	523·4
141	285·8	175	347·0	208	406·4	241	465·8	274	525·2
142	287·6	176	348·8	209	408·2	242	467·6	275	527·0
143	289·4	177	350·6	210	410·	243	469·4	276	528·8
144	291·2	178	352·4	211	411·8	244	471·2	277	530·6
145	293·0	179	354·2	212	413·6	245	473·0	278	532·4
146	294·8	180	356·0	213	415·4	246	474·8	279	534·2
147	296·6	181	357·8	214	417·2	247	476·6	280	536·0
148	298·4	182	359·6	215	419·0	248	478·4	281	537·8
149	300·2	183	361·4	216	420·8	249	480·2	282	539·6
150	302·0	184	363·2	217	422·6	250	482·0	283	541·4
151	303·8	185	365·0	218	424·4	251	483·8	284	543·2
152	305·6	186	366·8	219	426·2	252	485·6	285	545·0
153	307·4	187	368·6	220	428·0	253	487·4	286	546·8
154	309·2	188	370·4	221	429·8	254	489·2	287	548·6
155	311·0	189	372·2	222	431·6	255	491·0	288	550·4
156	312·8	190	374·0	223	433·4	256	492·8	289	552·2
157	314·6	191	375·8	224	435·2	257	494·6	290	554·0
158	316·4	192	377·6	225	437·0	258	496·4	291	555·8
159	318·2	193	379·4	226	438·8	259	498·2	292	557·6
160	320·	194	381·2	227	440·6	260	500·	293	559·4
161	321·8	195	383·0	228	442·4	261	501·8	294	561·2
162	323·6	196	384·8	229	444·2	262	503·6	295	563·0
163	325·4	197	386·6	230	446·0	263	505·4	296	564·8
164	327·2	198	388·4	231	447·8	264	507·2	297	566·6
165	329·0	199	390·2	232	449·6	265	509·0	298	568·4
166	330·8	200	392·	233	451·4	266	510·8	299	570·2
167	332·6	201	393·8	234	453·2	267	512·6	300	572·0
168	334·4								

TABLE III.*

SHOWING THE PRESSURE OF AQUEOUS VAPOUR IN INCHES OF MERCURY AT THE LATITUDE 53° 21′ FOR EACH DEGREE FAHRENHEIT FROM – 30° TO 432°.

Tempera-ture.	Pressure in Inches of Mercury at 32° at Sea Level.	Tempera-ture.	Pressure in Inches of Mercury at 32° at Sea Level.	Tempera-ture.	Pressure in Inches of Mercury at 32° at Sea Level.
°		°		°	
–30	0·0099	21	0·1121	71	0·7580
–29	0·0105	22	0·1171	72	0·7841
–28	0·0111	23	0·1223	73	0·8109
–27	0·0117	24	0·1278	74	0·8386
–26	0·0123	25	0·1335	75	0·8671
–25	0·0130	26	0·1395	76	0·8964
–24	0·0137	27	0·1457	77	0·9266
–23	0·0144	28	0·1522	78	0·9577
–22	0·0152	29	0·1589	79	0·9898
–21	0·0160	30	0·1660	80	1·0227
–20	0·0168	31	0·1733	81	1·0566
–19	0·0177	32	0·1810	82	1·0915
–18	0·0186	33	0·1883	83	1·1274
–17	0·0196	34	0·1959	84	1·1643
–16	0·0206	35	0·2038	85	1·2023
–15	0·0216	36	0·2119	86	1·2413
–14	0·0227	37	0·2204	87	1·2815
–13	0·0238	38	0·2291	88	1·3228
–12	0·0250	39	0·2381	89	1·3652
–11	0·0262	40	0·2475	90	1·4088
–10	0·0275	41	0·2571	91	1·4537
– 9	0·0289	42	0·2672	92	1·4998
– 8	0·0303	43	0·2775	93	1·5471
– 7	0·0317	44	0·2882	94	1·5958
– 6	0·0332	45	0·2993	95	1·6457
– 5	0·0348	46	0·3108	96	1·6971
– 4	0·0365	47	0·3226	97	1·7498
– 3	0·0382	48	0·3349	98	1·8039
– 2	0·0400	49	0·3476	99	1·8595
– 1	0·0419	50	0·3607	100	1·917
0	0·0439	51	0·3742	101	1·975
1	0·0459	52	0·3882	102	2·035
3	0·0503	53	0·4026	103	2·097
4	0·0526	54	0·4175	104	2·160
5	0·0551	55	0·4329	105	2·225
6	0·0576	56	0·4488	106	2·292
7	0·0603	57	0·4653	107	2·360
8	0·0630	58	0·4822	108	2·430
9	0·0659	59	0·4997	109	2·502
10	0·0689	60	0·5178	110	2·576
11	0·0721	61	0·5364	111	2·652
12	0·0753	62	0·5556	112	2·729
13	0·0788	63	0·5755	113	2·809
14	0·0823	64	0·5959	114	2·890
15	0·0861	65	0·6170	115	2·974
16	0·0899	66	0·6388	116	3·059
17	0·0940	67	0·6612	117	3·147
18	0·0982	68	0·6843	118	3·237
19	0·1027	69	0·7081	119	3·329
20	0·1073	70	0·7327	120	3·423

* Copied from the Rev. R. Dixon's 'Treatise on Heat.'

TABLE III.—*continued.*

Temperature.	Pressure in Inches of Mercury at 32° at Sea Level.	Temperature.	Pressure in Inches of Mercury at 32° at Sea Level.	Temperature.	Pressure in Inches of Mercury at 32° at Sea Level.
°		°		°	
121	3·520	173	13·033	225	38·50
122	3·619	174	13·333	226	39·23
123	3·720	175	13·639	227	39·98
124	3·824	176	13·951	228	40·74
125	3·930	177	14·268	229	41·52
126	4·039	178	14·592	230	42·30
127	4·150	179	14·922	231	43·10
128	4·264	180	15·258	232	43·91
129	4·381	181	15·600	233	44·73
130	4·500	182	15·949	234	45·57
131	4·622	183	16·304	235	46·42
132	4·747	184	16·666	236	47·28
133	4·874	185	17·034	237	48·15
134	5·005	186	17·410	238	49·04
135	5·139	187	17·792	239	49·94
136	5·275	188	18·181	240	50·85
137	5·415	189	18·577	241	51·78
138	5·558	190	18·981	242	52·72
139	5·704	191	19·392	243	53·67
140	5·854	192	19·810	244	54·64
141	6·006	193	20·236	245	55·63
142	6·162	194	20·669	246	56·62
143	6·322	195	21·110	247	57·64
144	6·485	196	21·559	248	58·66
145	6·651	197	22·016	249	59·71
146	6·822	198	22·480	250	60·76
147	6·996	199	22·953	251	61·84
148	7·173	200	23·435	252	62·92
149	7·354	201	23·924	253	64·03
150	7·540	202	24·422	254	65·15
151	7·729	203	24·929	255	66·28
152	7·922	204	25·445	256	67·43
153	8·120	205	25·969	257	68·60
154	8·321	206	26·502	258	69·79
155	8·527	207	27·045	259	70·99
156	8·737	208	27·597	260	72·20
157	8·951	209	28·158	261	73·44
158	9·170	210	28·728	262	74·69
159	9·393	211	29·308	263	75·96
160	9·621	212	29·898	264	77·24
161	9·853	213	30·50	265	78·55
162	10·090	214	31·11	266	79·87
163	10·332	215	31·73	267	81·21
164	10·579	216	32·35	268	82·56
165	10·831	217	32·99	269	83·94
166	11·088	218	33·64	270	85·33
167	11·350	219	34·30	271	86·75
168	11·617	220	34·98	272	88·18
169	11·889	221	35·66	273	89·63
170	12·167	222	36·35	274	91·10
171	12·450	223	37·05	275	92·59
172	12·739	224	37·77	276	94·10

TABLE III.—*continued.*

Temperature.	Pressure in Inches of Mercury at 32° at Sea Level.	Temperature.	Pressure in Inches of Mercury at 32° at Sea Level.	Temperature.	Pressure in Inches of Mercury at 32° at Sea Level.
°		°		°	
277	95·63	329	207·49	381	404·09
278	97·18	330	210·36	382	408·92
279	98·75	331	213·27	383	413·81
280	100·34	332	216·21	384	418·73
281	101·95	333	219·18	385	423·71
282	103·58	334	222·18	386	428·73
283	105·23	335	225·21	387	433·79
284	106·91	336	228·28	388	438·90
285	108·60	337	231·38	389	444·06
286	110·32	338	235·51	390	449·26
287	112·05	339	237·68	391	454·51
288	113·81	340	240·88	392	459·80
289	115·60	341	244·12	393	465·15
290	117·40	342	247·38	394	470·54
291	119·23	343	250·69	395	475·98
292	121·08	344	254·02	396	481·46
293	122·95	345	257·39	397	487·00
294	124·85	346	260·80	398	492·58
295	126·77	347	264·24	399	498·22
296	128·71	348	267·72	400	503·90
297	130·68	349	271·23	401	509·63
298	132·67	350	274·78	402	515·41
299	134·68	351	278·37	403	521·24
300	136·72	352	281·99	404	527·12
301	138·79	353	285·65	405	533·06
302	140·88	354	289·35	406	539·04
303	142·99	355	293·08	407	545·07
304	145·13	356	296·85	408	551·16
305	147·30	357	300·66	409	557·30
306	149·49	358	304·51	410	563·48
307	151·70	359	308·39	411	569·73
308	153·95	360	312·32	412	576·02
309	156·22	361	316·28	413	582·36
310	158·51	362	320·28	414	588·76
311	160·83	363	324·32	415	595·22
312	163·18	364	328·40	416	601·72
313	165·56	365	332·52	417	608·28
314	167·97	366	336·68	418	614·90
315	170·40	367	340·88	419	621·56
316	172·86	368	345·13	420	628·29
317	175·35	369	349·41	421	635·07
318	177·86	370	353·73	422	641·90
319	180·41	371	358·10	423	648·79
320	182·98	372	362·50	424	655·73
321	185·59	373	366·95	425	662·73
322	188·22	374	371·44	426	669·79
323	190·88	375	375·98	427	676·90
324	193·57	376	380·55	428	684·07
325	196·29	377	385·17	429	691·30
326	199·05	378	389·84	430	698·58
327	201·83	379	394·54	431	705·92
328	204·64	380	399·29	432	713·32

TABLE IV.

The tension in pounds is reduced from the tension in inches of mercury given in Table III.

1 inch mercury = ·48997 lb. 1 lb. water = 27·6933 cubic inches.

Barometer = 14·7 lbs. per square inch = 30·0018 inches.

$$S\,v = \frac{\cdot 0649\,T + 57\cdot 42}{\cdot 00299 \times P}$$, where P = tension in lbs. per square inch.

Deg. Fahr.	Tension of Aqueous Vapour in Lbs. per Sq. In.	Specific Volume.	Boiler Pressure.	Weight of a Cubic Foot of Steam in Lbs.	Deg. Fahr.	Tension of Aqueous Vapour in Lbs. per Sq. In.	Specific Volume.	Boiler Pressure.	Weight of a Cubic Foot of Steam in Lbs.
100°	·93927	22757	..	·0027423	146°	3·3426	6693	..	·0093236
101	·96769	22111	..	·0028224	147	3·4278	6533	..	·0095520
102	·99709	21481	..	·0029052	148	3·5146	6378	..	·0097846
103	1·0275	20866	..	·0029908	149	3·6032	6227	..	·010021
104	1·0584	20276	..	·0030778	150	3·6944	6080	..	·010265
105	1·0902	19706	..	·9031668	151	3·7870	5937	..	·01051
106	1·123	19149	..	·0032589	152	3·8816	5796	..	·01077
107	1·1563	18660	..	·0033444	153	3·9786	5661	..	·01103
108	1·1906	18099	..	·0034481	154	4·0770	5530	..	·01129
109	1·2259	17595	..	·0035467	155	4·1780	5402	..	·01155
110	1·2622	17105	..	·0036481	156	4·2809	5278	..	·01183
111	1·2994	16634	..	·0037518	157	4·3858	5155	..	·01210
112	1·3371	16181	..	·0038568	158	4·4930	5037	..	·01239
113	1·3763	15735	..	·0039659	159	4·6023	4923	..	·01268
114	1·4160	15310	..	·0040762	160	4·7140	4810	..	·01298
115	1·4572	14892	..	·0041906	161	4·8276	4702	..	·01327
116	1·4988	14493	..	·004306	162	4·9439	4595	..	·01358
117	1·5419	14134	..	·0044152	163	5·0624	4493	..	·01389
118	1·5860	13723	..	·0045474	164	5·1834	4392	..	·01421
119	1·6311	13357	..	·004672	165	5·3067	4294	..	·01453
120	1·6772	13003	..	·0047993	166	5·4328	4198	..	·01487
121	1·7247	12658	..	·0049303	167	5·5612	4105	..	·01520
122	1·7732	12324	..	·0050639	168	5·6789	4024	..	·01551
123	1·8227	12001	..	·0052000	169	5·8118	3935	..	·01586
124	1·8737	11659	..	·0053526	170	5·9478	3849	..	·01622
125	1·9256	11382	..	·0054828	171	6·1001	3757	..	·01661
126	1·9790	11086	..	·0056293	172	6·2274	3683	..	·01694
127	2·0333	10800	..	·0057780	173	6·3711	3604	..	·01732
128	2·0892	10522	..	·0059310	174	6·5328	3518	..	·01774
129	2·1466	10251	..	·0060879	175	6·6827	3442	..	·01813
130	2·2049	9989	..	·0062471	176	6·8356	3368	..	·01853
131	2·2645	9736	..	·0064096	177	6·9909	3297	..	·01893
132	2·3259	9489	..	·0065770	178	7·1496	3226	..	·01934
133	2·3881	9250	..	·0067462	179	7·3113	3158	..	·01976
134	2·4523	9017	..	·0069208	180	7·4760	3091	..	·02019
135	2·5180	8791	..	·0070992	181	7·6435	3027	..	·02062
136	2·5846	8572	..	·0072799	182	7·8145	2963	..	·02106
137	2·6532	8359	..	·0074658	183	7·9885	2902	..	·02151
138	2·7232	8152	..	·0076553	184	8·1659	2840	..	·02197
139	2·7948	7951	..	·0078489	185	8·3462	2782	..	·02243
140	2·8683	7755	..	·0080474	186	8·5304	2725	..	·02290
141	2·9428	7566	..	·0082484	187	8·7175	2669	..	·02338
142	3·0192	7382	..	·0084542	188	8·9081	2614	..	·02387
143	3·0976	7202	..	·0086653	189	9·1022	2561	..	·02437
144	3·1774	7028	..	·0088799	190	9·3001	2508	..	·02488
145	3·2588	6875	..	·0090776	191	9·5015	2458	..	·02539

TABLE IV.—*continued.*

Deg. Fahr.	Tension of Aqueous Vapour in Lbs. per Sq. In.	Specific Volume.	Boiler Pressure.	Weight of a Cubic Foot of Steam in Lbs.	Deg. Fahr.	Tension of Aqueous Vapour in Lbs. per Sq. In.	Specific Volume.	Boiler Pressure.	Weight of a Cubic Foot of Steam in Lbs.
° 192	9·7063	2408	..	·02592	° 244	26·772	915·2	12·072	·06818
193	9·8922	2365	..	·02639	245	27·257	899·6	12·557	·06938
194	10·127	2311	..	·02700	246	27·742	884·8	13·042	·07053
195	10·343	2267	..	·02753	247	28·242	869·9	13·542	·07174
196	10·563	2221	..	·02809	248	28·742	855·6	14·042	·07294
197	10·787	2176	..	·02868	249	29·256	840·7	14·556	·07423
198	11·014	2135	..	·02924	250	29·771	827·3	15·071	·07543
199	11·246	2091	..	·02984	251	30·299	813·6	15·599	·07670
200	11·483	2051	..	·03043	252	30·829	800·3	16·129	·07798
201	11·722	2011	..	·03104	253	31·373	787·3	16·673	·07927
202	11·966	1971	..	·03167	254	31·922	774·4	17·222	·08058
203	12·214	1933	..	·03227	255	32·476	761·7	17·776	·08193
204	12·467	1895	..	·03293	256	33·039	749·5	18·339	·08326
205	12·724	1859	..	·03356	257	33·612	737·4	18·912	·08463
206	12·985	1824	..	·03421	258	34·195	725·2	19·495	·08605
207	13·251	1789	..	·03489	259	34·783	713·8	20·083	·08743
208	13·522	1755	..	·03557	260	35·376	702·4	20·676	·08885
209	13·797	1720	..	·03628	261	35·984	691·2	21·284	·09028
210	14·076	1688	..	·03698	262	36·596	680·0	21·896	·09177
211	14·360	1656	..	·03768	263	37·218	669·3	22·518	·09323
212	14·649	1625	..	·03840	264	37·845	658·7	23·145	·09474
213	14·944	1595	·244	·03913	265	38·487	648·4	23·787	·09624
214	15·243	1565	·543	·03988	266	39·134	638·3	24·434	·09777
215	15·547	1535	·847	·04066	267	39·790	628·3	25·090	·09932
216	15·851	1507	1·151	·04140	268	40·452	618·5	25·752	·1009
217	16·164	1480	1·464	·04217	269	41·128	608·9	26·428	·1025
218	16·483	1453	1·783	·04297	270	41·809	599·5	27·109	·1041
219	16·806	1425	2·106	·04379	271	42·505	590·1	27·805	·1057
220	17·139	1399	2·439	·04460	272	43·205	581·1	28·505	·1074
221	17·472	1374	2·772	·04543	273	43·916	572·2	29·216	·1091
222	17·811	1349	3·111	·04626	274	44·636	563·4	29·936	·1108
223	18·153	1325	3·453	·04711	275	45·366	554·8	30·666	·1125
224	18·506	1300	3·806	·04800	276	46·106	546·4	31·406	·1142
225	18·864	1277	4·164	·04886	277	46·856	538·2	32·156	·1160
226	19·222	1255	4·522	·04975	278	47·615	530·0	32·915	·1177
227	19·589	1232	4·889	·05066	279	48·384	522·1	33·684	·1195
228	19·961	1210	5·261	·05157	280	49·164	514·2	34·464	·1214
229	20·344	1189	5·644	·05251	281	49·952	506·6	35·252	·1232
230	20·726	1165	6·026	·05356	282	50·634	500·2	35·934	·1248
231	21·118	1147	6·418	·05442	283	51·559	491·6	36·859	·1269
232	21·515	1126	6·815	·05540	284	52·382	484·3	37·682	·1289
233	21·916	1107	7·216	·05638	285	53·210	477·2	38·510	·1308
234	22·328	1088	7·628	·05738	286	54·054	470·2	39·354	·1327
235	22·744	1069	8·044	·05839	287	54·901	463·3	40·201	·1347
236	23·116	1050	8·466	·05944	288	55·763	456·5	41·063	·1367
237	23·592	1032	8·892	·06046	289	56·640	449·8	41·940	·1387
238	24·028	1014	9·328	·06153	290	57·522	443·3	42·822	·1408
239	24·469	996·8	9·769	·06261	291	58·419	436·9	43·719	·1428
240	24·915	979·8	10·215	·06370	292	59·326	430·5	44·626	·1450
241	25·371	963·1	10·671	·06479	293	60·241	424·4	45·541	·1470
242	25·831	946·9	11·131	·06590	294	61·172	418·3	46·472	·1492
243	26·242	932·9	11·542	·06690	295	62·114	412·3	47·414	·1513

TABLE IV.—*continued.*

Deg. Fahr.	Tension of Aqueous Vapour in Lbs. per Sq. In.	Specific Volume.	Boiler Pressure.	Weight of a Cubic Foot of Steam in Lbs.	Deg. Fahr.	Tension of Aqueous Vapour in Lbs. per Sq. In.	Specific Volume.	Boiler Pressure.	Weight of a Cubic Foot of Steam in Lbs.
296°	63·064	406·4	48·364	·1536	349°	132·89	201·5	118·19	·3097
297	63·882	401·6	49·182	·1554	350	134·63	199·1	119·93	·3134
298	65·004	395·0	50·304	·1580	351	136·39	196·6	121·69	·3174
299	65·989	389·3	51·289	·1603	352	138·17	194·2	123·47	·3213
300	66·989	383·9	52·289	·1626	353	139·96	191·9	125·26	·3252
301	68·003	378·5	53·303	·1649	354	141·77	189·6	127·07	·3291
302	69·027	373·2	54·327	·1672	355	143·60	187·4	128·90	·3330
303	70·061	368·0	55·361	·1696	356	145·45	185·1	130·75	·3372
304	71·109	362·9	56·409	·1720	357	147·31	183·0	132·61	·3411
305	72·173	357·8	57·473	·1744	358	149·2	180·8	134·50	·3452
306	73·246	352·8	58·546	·1769	359	151·1	178·7	136·40	·3493
307	74·322	348·0	59·622	·1793	360	153·03	176·6	138·33	·3534
308	75·431	343·2	60·731	·1818	361	154·61	174·9	139·91	·3568
309	76·543	338·5	61·843	·1843	362	156·93	172·5	142·23	·3618
310	77·665	333·9	62·965	·1869	363	158·91	170·4	144·21	·3661
311	78·802	329·4	64·102	·1895	364	160·91	168·4	146·21	·3705
312	79·953	324·9	65·253	·1921	365	162·92	166·5	148·22	·3747
313	81·120	320·5	66·420	·1947	366	164·96	164·6	150·26	·3793
314	82·301	316·2	67·601	·1974	367	167·02	162·7	152·32	·3836
315	83·491	311·9	68·791	·2001	368	169·11	160·8	154·41	·3881
316	84·696	307·7	69·996	·2028	369	171·20	159·0	156·50	·3926
317	85·916	303·6	71·216	·2055	370	173·32	157·1	158·62	·3971
318	86·945	300·2	72·245	·2078	371	175·46	155·3	160·76	·4018
319	88·396	295·6	73·696	·2111	372	177·61	153·6	162·91	·4063
320	89·655	291·7	74·955	·2140	373	179·79	151·8	165·09	·4110
321	90·934	287·8	76·234	·2168	374	181·99	150·1	167·29	·4157
322	92·222	284·0	77·522	·2197	375	184·22	148·4	169·52	·4204
323	93·525	280·3	78·825	·2226	376	186·46	146·7	171·76	·4253
324	94·844	276·6	80·144	·2256	377	188·72	145·2	174·02	·4300
325	96·176	273·0	81·476	·2286	378	191·02	143·5	176·32	·4349
326	97·529	269·5	82·829	·2316	379	193·32	141·9	178·62	·4398
327	98·891	266·0	84·191	·2346	380	195·64	140·3	180·94	·4447
328	100·27	262·5	85·57	·2378	381	197·99	138·8	183·29	·4497
329	101·66	259·0	86·96	·2409	382	200·36	137·2	185·66	·4548
330	103·07	255·8	88·37	·2440	383	202·76	135·7	188·06	·4599
331	104·49	252·5	89·79	·2471	384	205·17	134·2	190·47	·4650
332	105·94	249·4	91·24	·2502	385	207·61	132·8	192·91	·4700
333	107·39	246·1	92·69	·2536	386	210·06	131·3	195·36	·4754
334	108·86	242·9	94·16	·2569	387	212·54	129·9	197·84	·4804
335	110·35	239·8	95·65	·2602	388	215·05	128·4	200·35	·4859
336	111·85	236·8	97·15	·2635	389	217·58	127·1	202·88	·4912
337	113·37	233·8	98·67	·2669	390	220·12	125·7	205·42	·4964
338	114·9	231·0	100·20	·2702	391	222·7	124·4	208·00	·5018
339	116·46	228·0	101·76	·2737	392	225·29	123·0	210·59	·5074
340	118·04	225·3	103·34	·2770	393	227·91	121·7	213·21	·5128
341	119 61	222·5	104·91	·2805	394	230·55	120·4	215·85	·5185
342	121·21	219·7	106·51	·2840	395	233·22	119·1	218·52	·5227
343	122·83	217·0	108·13	·2876	396	235·9	117·9	221·20	·5296
344	124·46	214·2	109·76	·2913	397	238·62	116·6	223·92	·5352
345	126·11	211·7	111·41	·2948	398	241·35	115·3	226·65	·5411
346	127·78	209·0	113·08	·2985	399	244·11	114·1	229·41	·5467
347	129·47	206·5	114·77	·3023	400	246·9	112·9	232·20	·5525
348	131·18	204·0	116·48	·3060					

TABLE V.

Table Showing the Total Heat of Steam calculated according to Regnault's Formula $\lambda = 1081 \cdot 94 + \cdot 305$ T.

T is the number of degrees counted from 0° Fahr. The total heat at any temperature T is the amount necessary to turn 1 lb. of water at 32° Fahr. into saturated steam at the temperature T. The latent heat is the total heat as given in Column 2, minus the number of degrees above freezing at which the observation is made, see p. 113.

Deg. Fahr.	Total Heat.	Latent Heat.	Deg. Fahr.	Total Heat.	Latent Heat.	Deg. Fahr.	Total Heat.	Latent Heat.
40°	1094·14	1086·14	180°	1136·84	988·84	320°	1179·54	891·54
50	1097·19	1079·19	190	1139·89	981·89	330	1182·59	884·59
60	1100·24	1072·24	200	1142·94	974·94	340	1185·64	877·64
70	1103·29	1065·29	210	1145·99	967·99	350	1188·69	870·69
80	1106·34	1058·34	220	1149·04	961·04	360	1191·74	863·74
90	1109·39	1051·39	230	1152·09	954·09	370	1194·79	856·79
100	1112·44	1044·44	240	1155·14	947·14	380	1197·84	849·84
110	1115·49	1037·49	250	1158·19	940·19	390	1200·89	842·89
120	1118·54	1030·54	260	1161·24	933·24	400	1203·94	835·94
130	1121·59	1023·59	270	1164·29	926·29	410	1206·99	828·99
140	1124·64	1016·64	280	1167·34	919·34	420	1210·04	822·04
150	1127·69	1009·69	290	1170·39	912·39	430	1213·09	815·09
160	1130·74	1002·74	300	1173·44	905·44			
170	1133·79	995·79	310	1176·49	898·49			

Add for 1 degree	·305	Add for 1 tenth of a degree	·0305
„ 2 degrees	·610	„ 2 tenths „	·0610
„ 3 „	·915	„ 3 „ „	·0915
„ 4 „	1·22	„ 4 „ „	·122
„ 5 „	1·525	„ 5 „ „	·1525
„ 6 „	1·83	„ 6 „ „	·183
„ 7 „	2·135	„ 7 „ „	·2135
„ 8 „	2·44	„ 8 „ „	·244
„ 9 „	2·745	„ 9 „ „	·2745

1 millimetre = ·03937 English inch.
1 metre = 39·37079 „ inches.*
1 „ = 3·2809 „ feet.

$\frac{1}{7000}$ pound avoirdupois = 1 grain = $\frac{1}{5760}$ pound troy.

1 gramme = 15·43235 „ grains.†

1 cubic inch distilled water in a vacuum at 60° Fahr. weighs 252·769 grains ‡ = ·0361098 lb. avoir.

At 62° Fahr. in a vacuum, specific gravity of mercury, when water = 1, is 13·56889.§

1 cubic inch mercury at 60° Fahr. in a vacuum weighs 3429·797 grains = ·48997 lb. avoir.

30 inches mercury = 14·699 lbs. avoir.

1 lb. pressure per square inch equals 2·04094 inches mercury.

1 inch mercury = ·48997 lb. per square inch.

1 lb. of distilled water in a vacuum equals 27·6933 cubic inches.

1 litre = 61·02705 cubic inches. ‖

		Grains.
100 cubic inches dry air at 32° Fahr. under 30 inches mercury at level of sea at the equator ..		= 32·690541¶
„ „ Hydrogen	„ „	= 2·264147
„ „ Nitrogen	„ „	= 31·754611
„ „ Oxygen	„ „	= 36·143643
„ „ Carbonic acid	„ „	= 49·984164

Ratio of weight of mercury to weight of air at 32° Fahr., and under a pressure of 29·922 inches of mercury in latitude 45° is 10517·3 to 1.

* Balfour Stewart's 'Elementary Treatise on Heat,' 2nd Edition, p. 72. † Ibid. p. 74. ‡ Ibid. p. 75. § Ibid. p. 76. ‖ Dixon, p. 250. ¶ Ibid. p. 278, calculated from Regnault's determinations.

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